TRANSACTIONS

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

VOLUME 39

THIRTY-NINTH ANNUAL MEETING
CINCINNATI, OHIO, JANUARY 23-25, 1933

THIRTY-NINTH SEMI-ANNUAL MEETING
DETROIT, MICHIGAN, JUNE 22-24, 1933

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TRANSACTIONS

of

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

No. 941

THIRTY-NINTH ANNUAL MEETING, 1933

OUTSTANDING in the record of the 39th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS held at the Hotel Gibson, Cincinnati, January 23-25, were these notable facts: the intense interest in the subject matter of the technical papers and reports, the prompt adoption of the Revised Constitution and By-Laws, the action taken to create a committee to modernize nomenclature on heat output, and the large registration and attendance at all meetings and social functions.

President F. B. Rowley, Minneapolis, declared the 39th Annual Meeting of the Society in session at 9:30 A. M. Monday, January 23, at the Hotel Gibson, Cincinnati, Ohio.

Kenneth A. Wright, President of the Cincinnati Chapter, expressed his appreciation for the opportunity which the local members had to entertain the Society and then presented the Honorable Anthony B. Dunlap, a member of the City Council, who welcomed the Society members and guests to Cincinnati. A brief and fitting response was made by President Rowley.

The first item of business was the report of Council which was read by the Secretary.

Report of Council

During the year 1932, the Council held five regular meetings. The organization meeting was held in Cleveland, Ohio, at the conclusion of the 38th Annual Meeting, January 29, with President Rowley presiding. A. V. Hutchinson was appointed Secretary and the personnel of the four Council Committees was announced and confirmed by vote of the Council. Eight special committees were appointed.

The budget for 1932 as proposed by the Finance Committee was adopted providing for an estimated income of $83,420.00, an expenditure of $75,180.00 for all Society activities except Research.

At the April Meeting in Pittsburgh a vacancy in the Committee on Research was filled by the appointment of C. F. McIntosh to serve during 1932. A Charter was

granted for the formation of a Chapter in Cincinnati and a future schedule of Society meetings was discussed. Plans for the Semi-Annual Meeting 1932 in Milwaukee were approved.

During the June Meeting in Milwaukee a Charter was granted for the formation of a Student Chapter at the New York University. The budget was revised and a report from the Committee on Revision of Constitution and By-Laws was received. The publication of Transactions 1931 was deferred and a resolution was adopted proposing a reduction in dues for 1933 and 1934. Candidates were nominated for Committee on Research and a plan was proposed for the investigation of members delinquent on dues.

In September the Council met in Buffalo, New York, and received the report of the Committee on Revision of Constitution and By-Laws and authorized its preparation for final submission to the membership. The matter of membership and Society finances were discussed in detail.

On November 21 the Council met in New York and authorized the plan of limited chapter membership for the Massachusetts Chapter and Kansas City Chapter. Printing and distribution of the report of the Committee on Constitution and By-Laws was authorized so that action could be taken at the Annual Meeting in January, 1933.

The Council set the dues for 1933 on the following schedule: Members and Associates $18; Juniors $10; Students $3 and waived Initiation Fees for those elected during 1933.

The program for the 39th Annual Meeting in Cincinnati was approved.

During the year routine action was taken in the granting of Life Membership to 15 members; reinstatement of 8 members; acceptance of 121 resignations and cancellation of 142 memberships for non-payment of dues.

The report of the Committee on Increase of Membership was given by R. H. Carpenter, Chairman.

Report of Committee on Increase in Membership

The new Membership Drive, like everything else, suffered from existing business conditions during the year 1932. The methods followed by the Committee this year were somewhat different from those in existence previously. The results, while falling far short of the goal set by the Committee, may be considered fair nevertheless.

A total of 117 applications was acted upon by the Council during the calendar year, of which 100 represents new members of various grades. The classification of same is as follows:

NEW APPLICATIONS		ADVANCEMENTS		RE-INSTATEMENTS	
Members	47	Members	9	Members	7
Associates	20			Associates	1
Juniors	15				
Students	18				
	100				

Respectfully submitted,
COMMITTEE ON INCREASE IN MEMBERSHIP,
R. H. CARPENTER, *Chairman.*

John Howatt, Chairman of the Finance Committee, gave a summary of the financial condition of the Society.

Report of the Finance Committee

The year 1932 was a year in which all corporations, societies and individuals that operated on a budget system experienced a shrinkage in revenue beyond that anticipated. The most pessimistic prophecies made at the beginning of 1932 proved to be

the most nearly correct. The experience of our Society was no different than that of other organizations. We prepared a budget at the beginning of the year, based upon what was at the time considered to be a reasonable reduction in revenue below that of 1931. By mid-year, however, it was evident that our anticipated revenues were not to be obtained, so at the summer meeting in Milwaukee, your Finance Committee prepared and had approved by the Council, a revised budget which greatly curtailed expenditures for the balance of the year. The expenses of the Society were maintained within the revised budget provisions and the year closed without a budget deficit. The shrinkage in both revenues and expenditures for the year 1932 as compared with 1931 was approximately 25 per cent.

As required by our Constitution and By-Laws, an examination of our funds and audit of our accounts was made by a Certified Public Accountant for the year ending December 31, 1932. A copy of his report is filed with the Secretary and may be examined by any member of the Society. The main points of interest in this report are as follows:

(1) There was cash on deposit in the various Society Funds totaling $10,280.34, on December 31, 1932.

(2) The Society owns securities of a par value of $36,000 which cost the Society $35,680.67 but which have a present market value of $20,272.50, a shrinkage of $15,408.17 or 43%. Up to date there has been no default in the interest payment on any security held by the Society.

(3) The General Fund, or Capital of the Society decreased from $34,224.87 to $32,823.93 during the year, due to a paper loss of $2,356.53 reported by the Auditor in all Society activities.

(4) The accounts receivable as of date December 31, 1932, were $28,513.84 consisting mainly of unpaid members' dues for 1930, 1931 and 1932, and monies due on advertising in the Guide and on sales of Guide copies.

(5) The accounts payable consist of $11,631.24, principally monies due on the production of the 1933 Guide and which will be paid out of the receipts from advertisements in the same Guide, and the sum of $7,500.00 set up as a contingent liability for the production of the 1931 and 1932 TRANSACTIONS. The total of accounts payable is $22,271.11 as contrasted with accounts receivable of $28,513.84.

(6) The gross income to the Society from all sources was $64,944.00 in 1932. The total expenses as shown by the Auditor were $67,300.53. This latter figure, however, includes an item of $3,500.00 for 1931 TRANSACTIONS which was not budgeted and on which nothing was actually spent, the sum of $1,357.21 to cover the loss on 1930 TRANSACTIONS, a loss carried over from previous years, and the sum of $1,500.00 set up as a possible loss in collection on advertising contracts. The Budget was followed closely throughout the year and the year closed without a Budget deficit.

Your Finance Committee has estimated the gross income for the Society for 1933 at approximately $54,000. In order to live within this income it will be necessary to again call on the employes at the Headquarters Office to operate on a reduced income basis, to reduce salaries, to reduce rent for office space and to curtail some of the Society activities. It is the sincere hope of the Committee that this is the last year in which your Finance Committee will be required to prepare a Budget based upon a reduced income or one that will show any serious curtailment of Society activities.

FINANCE COMMITTEE,
JOHN HOWATT, *Chairman.*

Report of Certified Public Accountant

January 8, 1933.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS,
51 MADISON AVENUE,
NEW YORK CITY.

Gentlemen:

Pursuant to your request I made an examination of the books of account and records of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, New York City, for the year ended December 31, 1932, and submit herewith my report.

The work covered a verification of the Assets and Liabilities as of the date previously stated and a review of the operating accounts for the Calendar Year 1932.

Submitted herewith is a Balance Sheet showing the financial condition of the Society on December 31, 1932, and your attention is directed to the following comments thereon:

CASH

Cash on Deposit was verified by direct communication with the banks listed below and by reconcilement of the amounts reported to me with the balances shown by the books of the Society.

SOCIETY

BANKS	AMOUNT
Chase National Bank (Special)	$1,092.75
Chase National Bank (Regular)	1,433.41
Bankers Trust Company	1,581.66
Bowery Savings Bank (Book No. 57,001)	935.99
Emigrant Industrial Savings Bank (Book No. 141,682)	1,411.31
Bank of United States in Liquidation No. 19,905	500.00
	$6,955.12

FUNDS

F. PAUL ANDERSON AWARD		
Bank for Savings (Book No. 1,326,641)	$ 706.04	
Bank of United States in Liq. No. 19,905	185.56	891.60
RESEARCH		
Bankers Trust Company	1,852.35	
Forbes National Bank, Pittsburgh, Pa	24.03	1,876.38
RESEARCH ENDOWMENT FUND		
Bank for Savings (Book No. 1,326,640)	$ 282.81	
Bank of United States in Liq. No. 19,906	274.43	557.24
TOTAL		$10,280.34

Cash on Hand for Deposit was verified by inspection of the checks or of the record made of Cash Receipts subsequent to December 31, 1932.

The Petty Cash at New York was verified by count.

MARKETABLE SECURITIES

There is attached hereto a schedule of negotiable bonds which were verified by direct communication with the Bankers Trust Company where same are deposited for safekeeping. No adjustment has been made of the $15,408.17 shrinkage in the market value of these securities. These have been included in the attached Balance Sheet at cost.

ACCOUNTS RECEIVABLE

Unpaid membership dues were determined by trial balance of the individual ledger cards and aging of the unpaid charges which may be summarized below:

1932 Unpaid Dues	$18,390.00
1931 Unpaid Dues	6,923.55
1930 Unpaid Dues	616.00
TOTAL	$25,929.55

My verification of the dues disclosed that during the year 1932 there had been prepaid to the Society dues amounting to $304.93 which sum I have shown on the attached Balance Sheet as deferred income.

The Reserve found on the books to cover probable losses which may result from prior years' dues was found ample. In addition, however, I have provided the sum of $12,657.38, which will increase the reserve to seventy-five (75%) per cent of the total dues outstanding. Adequate reserves have also been provided to cover losses which may be incurred during the realization of all other Accounts Receivable.

BALANCE SHEET

American Society of Heating and Ventilating Engineers

For the Year Ended December 31, 1932

ASSETS

SOCIETY				
Cash				
On Deposit		$ 6,955.12		
On Hand		100.00	$ 7,055.12	
Investments (at Cost)				
Securities (Market Value $9,-917.50)		12,919.49		
ADD: Accrued Interest		78.75	12,998.24	
Accounts Receivable				
Membership Dues	$25,929.55			
LESS: Reserve for Doubtful	19,493.66	6,435.89		
Advertisements	22,537.05			
LESS: Reserve for Doubtful	1,379.43	21,157.62		
Other	1,020.33			
LESS: Reserve for Doubtful	100.00	920.33	28,513.84	
Inventories				
Transactions 1925–1930		2,408.63		
Emblems and Certificate Frames		72.80		
Postage		445.00		
Stationery, Printing and Office Supplies		289.00	3,215.43	
Permanent				
Library		300.00		
Furniture and Fixtures	5,215.46			
LESS: Reserve for Depreciation	2,736.35	2,479.11	2,779.11	
Deferred Charges				
Meetings 1933		33.30		
Chapter Meetings Allowance		500.00	533.30	$55,095.04
SPECIFIC FUNDS				
Endowment Fund				
Securities at Cost (Market Value $7,775.00)		19,753.68		
ADD: Accrued Interest		250.83	20,004.51	
F. Paul Anderson Award Fund				
Cash on Deposit		891.60		
Cash on Hand for Deposit		152.34	1,043.94	
Research Fund				
Cash on Deposit—Research	1,876.38			
Cash on Hand for Deposit—Research	3,020.00			
Cash on Hand—Pittsburgh, Pa.	50.00			
Cash on Deposit—Research Endowment Fund	557.24			
Cash on Hand for Deposit—Research Endowment Fund	61.00	5,564.62		
Securities at Cost (Market Value $2,580.00)	3,007.50			
ADD: Accrued Interest	45.00	3,052.50	8,617.12	29,665.57
				$84,760.61

LIABILITIES AND CAPITAL

SOCIETY			
ACCOUNTS PAYABLE		$11,631.24	
DUE RESEARCH FUND		2,834.94	
RESERVE FOR TRANSACTIONS			
1931	$4,000.00		
1932	3,500.00	7,500.00	
DEFERRED INCOME			
Prepaid Dues		304.93	
		22,271.11	
TOTAL LIABILITIES			
GENERAL FUND			
Society		32,823.93	$55,095.04
SPECIFIC FUNDS			
Endowment		20,004.51	
F. Paul Anderson Award		1,043.94	
Research	7,998.88		
Research Endowment	618.24	8,617.12	29,665.57
			$84,760.61

Note "A": This Balance Sheet is subject to the comments contained in the letter attached to and forming a part of this report.

INVENTORIES

All TRANSACTIONS on hand for the years 1925 to 1930, inclusive, were inventoried and priced at cost.

TRANSACTIONS covering the year 1931, volume 37, and the year 1932, volume 38, were not published up to the close of the current year, therefore reserves of $4,000.00 and $3,500.00, respectively, have been provided to cover the future cost thereof.

All other Inventories were verified either by actual count or analysis of the records.

ACCOUNTS PAYABLE

On December 31, 1932, there remained unpaid invoices amounting to $11,631.24, of which $11,320.39 was owing on account of the Guide and Year Book printing costs.

DUE RESEARCH LABORATORIES

Of the dues charged to Members and Associates, forty (40%) per cent has been reserved for the Research Laboratories in accordance with Section 5, Article 3, of the By-Laws. The sum payable to the Research Laboratory as and when the dues receivable will have been realized in cash is $2,634.94. An analysis of the amount due the Research Laboratories follows:

Dues	$2,634.94	
Interest from Endowment Fund Securities	1,025.00	$ 3,659.94
Clerical Salaries and Custodian's Fee chargeable to Research		825.00
Balance Due Research		$ 2,834.94

GENERAL FUND

An analysis of the General Fund of the Society showing the changes made during the year under audit follows:

GENERAL FUND—January 1, 1932—PER FORMER REPORT	$34,224.87
Addition	
Expenses of the Committee on Research previously absorbed by the Society	955.59
	$35,180.46

Deductions

Loss from Society Activities for the Calendar Year 1932 from Society Statement of Income and Expenses	$ 180.01	
Loss from Guide for the Calendar Year 1932 from Guide Statement of Income and Expenses	2,176.52	2,356.53
GENERAL FUND—DECEMBER 31, 1932—PER BALANCE SHEET		$32,823.93

MEMBERSHIP

A comparison of the membership in force as of the close of business December 31, 1931 and 1932, respectively, follows:

CLASSIFICATION	1932	1931	INCREASES DECREASES
Members	1403	1539	136
Associates	424	490	66
Juniors	167	203	36
Students	44	22	22
Honorary	2	2	..
Life	27	25	2
	2067	2281	214

RESEARCH LABORATORIES

In accordance with Council Minutes the annual report of the Research Laboratories, which in prior years was rendered under separate cover, has been bound herein.

Respectfully submitted,
FRANK G. TUSA,
Certified Public Accountant.

BUDGET COMPARISON—SOCIETY ACTIVITIES

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, NEW YORK CITY

For the Year Ended December 31, 1932

	ACTUAL 1932	BUDGET PROVISION	INCREASES DECREASES
INCOME			
Current Dues—Less Reserve	$21,358.95	$20,500.00	$ 858.95
Initiation Fees	970.00	1,100.00	130.00
Sales—Emblems and Certificate Frames	104.00	100.00	4.00
Sales—Codes	46.47	75.00	28.53
Sales—Journals, Reprints and Books	388.86	300.00	88.86
Sales—Transactions	386.25	500.00	113.75
Editorial Contract (Net)	11,127.36	11,000.00	127.36
Interest Earned—General Funds	620.00	300.00	320.00
Interest Earned—Current Funds	253.71	200.00	53.71
	$35,255.60	$34,075.00	$ 1,180.60
EXPENSES			
Salaries (Exclusive of Secretary)	$ 9,214.53	$ 9,500.00	$ 285.47
Postage	2,477.36	3,000.00	522.64
General Printing	836.32	700.00	136.32
Yearbook	725.00	900.00	175.00
Reprints, Books and Journals	267.78	200.00	67.78
Meetings	2,555.99	2,500.00	55.99
Meetings—Council, etc	221.73	250.00	28.27
Traveling—President	734.92	900.00	165.08
Codes	722.44	400.00	322.44
Publicity	1,226.17	1,250.00	23.83
President's Emergency Fund	210.31	250.00	39.69
70% of Apportionable Expenses	11,175.38	11,893.00	717.62
	$30,367.93	$31,743.00	$ 1,375.07
UNBUDGETED EXPENSES			
Transactions—Prior Years	$ 1,357.21	....	$ 1,357.21
Transactions—1932	3,500.00	....	3,500.00
Engraving Certificates	100.35	....	100.35
Emblems and Certificate Frames	110.52	....	110.52
	$35,436.01	$31,743.00	$ 3,693.01
LOSS FOR THE YEAR	$ 180.41	$ 2,332.00	$ 2,512.41

Note "A": General Printing includes $287.84 expended to print the Constitution and By-Laws.

BUDGET COMPARISON—GUIDE

INCOME	ACTUAL	BUDGET PROVISION	INCREASES DECREASES
Advertising	$17,371.14	$29,000.00	$11,628.86
Sales	12,317.26	11,000.00	1,317.26
Totals	$29,688.40	$40,000.00	$10,311.60
EXPENSES			
Printing and Binding 1932 Issue	$ 8,834.15	$11,500.00	$ 2,665.85
Mailing 1932 Issue	2,679.62	2,500.00	179.62
Paper Purchases	1,553.27	2,400.00	846.73
Engraving and Art Work	428.60	500.00	71.40
Sales Promotion—Advertising	1,111.00	1,750.00	639.00
Sales Promotion—Copy Sales	3,137.56	4,000.00	862.44
Salaries (Exclusive of Secretary)	6,831.27	6,900.00	68.73
30% of Apportionable Expenses	4,789.45	5,097.00	307.55
Totals	$29,364.92	$34,647.00	$ 5,282.08
GUIDE PROFIT	$ 323.48	$ 5,353.00	$ 5,029.52
OTHER DEDUCTIONS			
Chapter Meeting Allowance	1,000.00	....	1,000.00
Provision for Doubtful Accounts	1,500.00	....	1,500.00
GUIDE LOSS	$ 2,176.52	$ 5,353.00	$ 7,529.52

A report of the Secretary was given by A. V. Hutchinson, and showed clearly the present condition of the Society.

Report of Secretary

The Society during 1932 carried on a record volume of activities and despite the drastic retrenchment program instituted, due to the reduction of revenue from dues and advertising, the major part of its contemplated program was carried out.

The TRANSACTIONS 1930 was distributed immediately after the Annual Meeting in Cleveland and several Codes were completed and sent to the members. These included:

A. S. H. V. E. Standard Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers.

A. S. H. V. E. Standard Code for Testing Steam Heating Boilers Burning Oil Fuel.

A. S. H. V. E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code).

A. S. H. V. E. Standard Code for Testing and Rating Steam Unit Ventilators.

Report of A. S. H. V. E. Committee on Ventilation Standards.

These projects resulted from the untiring work of committees assigned to produce them, some of which have been working through a period of several years and have just completed their labors.

THE GUIDE 1933 was compiled with the cooperation of assistants among a widely scattered group of Society members working with the Guide Publication Committee, directed by Mr. Boyden. During the year the sale of copies of the 1932 edition was unusually heavy and exceeded the total of the previous year.

When it was decided to hold the 39th Annual Meeting in Cincinnati, the local group of members organized a Chapter and regular meetings were started during the Fall.

Earlier in the year the Students of New York University petitioned the Council for a Charter and a successful Student Chapter was organized and has the honor of being the first Student Section.

The majority of Chapters were visited by President Rowley, who carried to them an inspiring message on the value of Society membership, the importance of the Research projects, which it sponsors and the recognition that it has received through the compilation of THE GUIDE and its codes and standards.

The collection of dues presented a very difficult problem for the Finance Committee

and revisions of the budget were necessary in the middle of the year. It should be noted, however, that the revised estimates of income and the actual expenditures were very accurate. With many members experiencing reduction of income and a considerable number suffering loss of employment, collection of dues has held up remarkably because of members' interest in and loyalty to the Society.

The effort to secure employment for members has continued and this has involved an unusually heavy correspondence at the Headquarters office. Recently an increasing inquiry for men trained in specialized phases of heating, ventilating and air conditioning has been noticeable.

Some loss of membership occurred during the past year and the present status of membership is 2,067, a reduction of 9 per cent from the peak reached in 1931. The loss by death has been unusually heavy and the total of 29 for 1932 equals the combined loss of the last 5 years.

Through the work of the Committee on Increase of Membership and the Council Members and Chapter Committees, a total of 117 men were elected during 1932. The present status of membership is as follows: Members 1,403, Associates 454, Juniors 137, Students 44, Life Members 27, Honorary Members 2.

The extra effort required to carry on the Society's work this year imposed by the financial conditions in the profession and industry, was supplied by the splendid energy and enthusiasm of my associates on the staff at the headquarters office. Each one ably met the daily assignments and gave extra hours without compensation as the personnel was reduced to meet income. Their idea was to bring the Society through without impairment of the service. The year 1933 starts with a 50 per cent reduction in personnel and a proportionately reduced budget. There is, however, a definite indication of a widespread interest in Society activities and the need of the coming year is for every member to get a member, so that the organization can carry on the fine work that has won for it widespread recognition.

Respectfully submitted,
A. V. HUTCHINSON, *Secretary.*

Report of the Committee on Research

The Report of the Committee on Research was prepared by G. L. Larson, Chairman, and F. C. Houghten, Director of the Research Laboratory, and was presented by Professor Larson.

THE Committee on Research of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS held four meetings during 1932, two during the Annual Meeting at Cleveland in January, one in Pittsburgh in April, and one at the time of the Semi-Annual Meeting in Milwaukee in June. Financial difficulties resulting from the depression, and the problem of scaling down activities to make expenditures come within diminishing receipts occupied a large part of the Committee's deliberations. In this connection it was found necessary to curtail some of the research activities and otherwise reduce expenditures. Funds to meet the budget were derived from dues of Society members, interest on reserve funds, proceeds from the Second International Heating and Ventilating Show in Cleveland, profits from the publication of THE GUIDE, and contributions from industry.

BUDGET 1932

Revised April 8, 1932*

INCOME	BUDGET 1932	ACTUAL 1932
A. S. H. V. E. Dues and Guide	$20,000.00	$16,795.92
Manufacturers	5,000.00	5,745.00
Association	3,000.00	3,525.00
Exposition	2,250.00	5,012.76
Other Contributions	5,000.00	2,500.00
Interest on Bank Balances	60.00	8.20
Interest and Securities	100.00	135.00
Interest on Society Endowment	1,000.00	1,025.00
Surplus Fund	1,750.00	....
	$38,160.00	$34,746.88

* Original budget adopted January 27, 1932.

EXPENSES	BUDGET 1932	ACTUAL 1932
Travel—Committee and Laboratory Personnel	$ 2,000.00	$ 887.75
Salary—Clerical, Society Headquarters	800.00	500.00
Salary—Technical Adviser	2,800.00	2,800.00
General Printing and Promotion	500.00	10.51
Correlating Thermal Research	500.00	500.00
A. S. H. V. E. Laboratory (Pittsburgh)		
Salaries	$15,000.00	$12,449.16
Laboratory Supplies and Equipment	1,500.00	571.71
Office Supplies and Expense	750.00	845.41
Meetings	200.00	67.25
Contracts with Cooperating Institutions	13,175.00	12,789.50
Emergency Fund (For use as directed by Committee on Research)	935.00	....
	$38,160.00	$31,421.29
Cleveland Exposition	....	432.34
Dispersed for Res. Lab. by Soc. Hdq. Off. 1931	....	850.00
Total Expense	$38,160.00	$32,703.63

1932 Research Contributors

American Oil Burner Association
Copper and Brass Research Association
Heating and Piping Contractors National Association
National Association of Fan Manufacturers
National Association of Ice Industries
Ventilating Contractors Employers Association
Aerofin Corporation
American Air Filter Company
American Blower Corporation
American Radiator Company
Barber-Colman Company
Barnes and Jones
Brundage Company
Buckeye Blower Company
A. M. Byers Company
Buffalo Forge Company
Bush Manufacturing Company
W. H. Carrier
Carrier Engineering Corporation
Celotex Company
Clarage Fan Company
Crane Company
Detroit Edison Company
Detroit Stoker Company
C. A. Dunham Company
Frigidaire Corporation
Garden City Fan Company
General Electric Company
Hart and Hutchinson Company
Hoffman Specialty Company
Holtzer-Cabot Electric Company
Howard Iron Works and Alberg Heater Company
Ilg Electric Ventilating Company
Illinois Engineering Company
Johns-Manville Corporation
Johnson Service Company
Leeds and Northrup Company
Modine Manufacturing Company
Minneapolis-Honeywell Regulator Company
Nash Engineering Company
National Radiator Corporation
National Regulator Company
National Tube Company
Herman Nelson Corporation
J. J. Nesbitt, Inc.
New York Blower Corporation
Owens-Illinois Glass Company
Peerless Unit Ventilation, Inc.
Petroleum Heat and Power Company
Powers Regulator Company
Reliable Plumbing and Heating Company
Smithsonian Institution
B. F. Sturtevant Company
Trane Company
United Engineers and Constructors
United States Radiator Corporation
Warren Webster and Company
Westinghouse Electric and Manufacturing Company
Youngstown Sheet and Tube Company

Research Activities

The Committee on Research gave consideration during 1932 to 14 research projects. While it was impossible to investigate each subject in the laboratory, a Technical Advisory Committee studied each project with a view of determining its needs and of cooperating with the Chairman of the Committee on Research and the Director of the Research Laboratory in developing plans for research to be carried on in the laboratory at Pittsburgh and in the cooperating universities. The fourteen Technical Advisory Committees, each made up of authorities on the particular subjects, brought to the active support of the Society's research activity the combined experience and

ability of 88 engineers, educators and scientists. During 1932 Prof. A. C. Willard continued as Technical Advisor to the Committee on Research.

Cooperating Universities

The practice of augmenting the work carried on at the Society's laboratory in Pittsburgh through cooperative arrangements with engineering departments of educational institutions was continued. The following is a list of institutions which cooperated with the Laboratory in 1932 and the projects under investigation by them:

University of Wisconsin: Aeration of buildings and allied subjects.

Armour Institute of Technology: Study of air flow through registers and grilles and performance of unit ventilators.

Carnegie Institute of Technology: Study of capacity of pipe for steam heating systems.

University of Illinois: Direct and indirect radiation with gravity air circulation. Also cooling of buildings.

University of Kansas: Garage ventilation.

Harvard School of Public Health: Ionization of air and its health significance.

Yale University: Study of oil burning devices.

University of Minnesota: Transmission of heat through building construction and allied subjects.

Agricultural and Mechanical College of Texas: Flow of water in hot water heating systems and allied subjects.

Papers Published Since the 1932 Annual Meeting

As a result of the activities of the Technical Advisory Committees, 15 papers were prepared for presentation at either the 1932 Summer Meeting or the 1933 Annual Meeting of the Society. Each of these papers added something to the fundamental technical knowledge of the heating, ventilating and air conditioning branches of the engineering profession. These papers are:

Thermal Properties of Building Materials, by F. B. Rowley and A. B. Algren. (A. S. H. V. E. TRANSACTIONS, Vol. 38, 1932). Presented at the Semi-Annual Meeting, June 1932.

Natural Wind Velocity Gradients Near a Wall, by J. L. Blackshaw and F. C. Houghten. (A. S. H. V. E. TRANSACTIONS, Vol. 38, 1932). Presented at the Semi-Annual Meeting, June 1932.

How to Use the Effective Temperature Index and Comfort Charts (Report of Technical Advisory Committee on Re-Study of Comfort Chart and Comfort Line, C. P. Yaglou, *Chairman*). (A. S. H. V. E. TRANSACTIONS, Vol. 38, 1932). Presented at the Semi-Annual Meeting, June 1932.

Investigation of Air Outlets in Class Room Ventilation, by G. L. Larson, D. W. Nelson and R. W. Kubasta. (A. S. H. V. E. TRANSACTIONS, Vol. 38, 1932). Presented at the Semi-Annual Meeting, June 1932.

Loss of Head in Copper Pipe and Fittings, by F. E. Giesecke and W. H. Badgett. (A. S. H. V. E. TRANSACTIONS, Vol. 38, 1932). Presented at the Semi-Annual Meeting, June 1932.

Carbon Monoxide Distribution in Relation to the Ventilation of a One-Floor Garage, by F. C. Houghten and Paul McDermott. (A. S. H. V. E. TRANSACTIONS, Vol. 38, 1932). Presented at the Semi-Annual Meeting, June 1932.

Carbon Monoxide Distribution in Relation to the Ventilation of an Underground Ramp Garage, by F. C. Houghten and Paul McDermott. (A. S. H. V. E. TRANSACTIONS, Vol. 38, 1932). Presented at the Semi-Annual Meeting, June 1932.

Test of Convectors in a Warm Wall Testing Booth, by A. P. Kratz, M. K. Fahnestock and E. L. Broderick. (A. S. H. V. E. TRANSACTIONS, Vol. 38, 1932). Presented at the Semi-Annual Meeting, June 1932.

Study of the Application of Thermocouples to the Measurement of Wall Surface Temperatures, by A. P. Kratz and E. L. Broderick. (A. S. H. V. E. TRANSACTIONS, Vol. 39, 1933). Presented at the Annual Meeting, January 1933.

Flow of Condensate and Air in Steam-Heating Returns, by F. C. Houghten and Carl Gutberlet. (A. S. H. V. E. TRANSACTIONS, Vol. 39, 1933). Presented at the Annual Meeting, January 1933.

Air Supply and Its Effect on Performance of Oil Burners and Heating Boilers, by L. E. Seeley, J. H. Powers and E. J. Tavanlar. (A. S. H. V. E. TRANSACTIONS, Vol. 39, 1933). Presented at the Annual Meeting, January 1933.

Condensate and Air Return in Steam Heating Systems, by F. C. Houghten and J. L. Blackshaw. (A. S. H. V. E. TRANSACTIONS, Vol. 39, 1933). Presented at the Annual Meeting, January 1933.

Study of Summer Cooling in the Research Residence at the University of Illinois, by A. P. Kratz and S. Konzo, and Summer Cooling Operating Results in a Detroit Residence, by J. H. Walker and G. B. Helmrich. (A. S. H. V. E. TRANSACTIONS, Vol. 39, 1933). Presented at the Annual Meeting, January 1933.

Cold Walls and Their Relation to the Feeling of Warmth, by F. C. Houghten and Paul McDermott. (A. S. H. V. E. TRANSACTIONS, Vol. 39, 1933). Presented at the Annual Meeting, January 1933.

Research Projects Considered During 1932

Each of the 14 research projects under consideration by the 14 Technical Advisory Committees are discussed in detail under the subject headings and the personnel of the committees.

1. **INFILTRATION IN BUILDINGS.**—*Technical Advisory Committee:* J. G. Shodron, *Chairman;* J. E. Emswiler, F. E. Giesecke, D. W. Nelson, W. C. Randall, Ernest Szekely, J. H. Walker, M. S. Wunderlich.

The general subject of aeration of buildings has been investigated by the Research Laboratory and cooperative universities for the past several years. During the past year the project has been continued at the University of Wisconsin in cooperation with the Society under the direction of Prof. G. L. Larson, chairman of the Mechanical Engineering Department of the University. The Technical Advisory Committee has outlined a number of phases of the general study still to be investigated.

Tests on the influence of outlet vents were reported at the June 1932 Meeting of the Society in a paper entitled, Investigation of Air Outlets in Class Room Ventilation, by G. L. Larson, D. W. Nelson and R. W. Kubasta. (See A. S. H. V. E. TRANSACTIONS, Vol. 38, 1932.)

A study of air stratification in confined spaces and the effect of crowds and air movement has been carried on for the last two winters in the new University Field House. Many series of temperature readings have been taken under various conditions of air circulation. Resistance thermometers are placed at 10-ft intervals from the floor to roof. Records of relative humidities have also been taken. The results without fan circulation show a fairly large temperature gradient in the first 25 ft and a much smaller gradient in the last 75 ft. The system is designed in such a way that the fan circulation reduces the temperature gradient to a very low amount.

2. **AIR FLOW THROUGH REGISTERS AND GRILLES.**—*Technical Advisory Committee:* John Howatt, *Chairman;* J. J. Aeberly, C. A. Booth, L. E. Davies, D. E. French, J. J. Haines.

The work on air flow through registers and grilles has been given consideration by the Technical Advisory Committee during the past year and work has been carried on by Prof. L. E. Davies at Armour Institute of Technology in cooperation with the Society. The work has been directed along two definite lines: first, the study of methods for the measurement of air flow through registers and grilles when being supplied to or exhausted from rooms; and second, the study of methods of determining air delivery by unit ventilators.

3. **PIPE AND TUBING CARRYING LOW PRESSURE STEAM OR HOT WATER.**—*Technical Advisory Committee:* S. R. Lewis, *Chairman;* J. C. Fitts, F. E. Giesecke, H. M. Hart, A. P. Kratz, C. A. Hill, R. R. Seeber, W. K. Simpson.

The study of pipe capacities for steam and hot water heating systems has been under investigation by the Laboratory for the past several years. The work on pipe sizes for steam heating systems was carried on by the Laboratory at the Carnegie Institute of Technology in cooperation with the latter institution.

These studies have resulted in the papers, Flow of Condensate and Air in Steam Heating Returns, by F. C. Houghten and Carl Gutberlet, and Condensate and Air Return in Steam Heating Systems, by F. C. Houghten and J. L. Blackshaw. (See A. S. H. V. E. TRANSACTIONS, Vol. 39, 1933.)

The study of the carrying capacity of pipe in hot water heating systems has been investigated by Prof. F. E. Giesecke, Director of the Experiment Station of the Texas Agricultural and Mechanical College in cooperation with the Society. This investigation has been under way for some time, resulting in a number of publications.

4. **GAS HEATING EQUIPMENT.**—*Technical Advisory Committee:* W. E. Stark, *Chairman;* R. M. Conner, Robert Harper, Thomson King, F. B. Howell, E. A. Jones, Henry Loebell, J. F. McIntire, L. S. Ourusoff, H. L. Whitelaw.

The Technical Advisory Committee on Gas Heating Equipment was organized during the present year and has undertaken a survey of the work to be carried on by the Laboratory. In this connection the committee has proposed a program including the following: (1) a study of the value of automatic draft adjusters on gas boilers with an estimate of the fuel saving that may be expected from their use;

(2) a study of the loss of heat through draft hoods under normal gas house heating operating conditions.

5. SOUND IN RELATION TO HEATING AND VENTILATION.—*Technical Advisory Committee:* Warren Ewald, *Chairman;* Carl Ashley, C. A. Booth, V. O. Knudsen, R. F. Norris, J. P. Reis, G. T. Stanton.

The Technical Advisory Committee on this subject has studied the needs of the problem during the current year but have not yet presented a comprehensive plan for laboratory research. The problem is a new one as far as the Society's research activity is concerned, and will no doubt prove a fruitful field of investigation for the future.

6. DIRECT AND INDIRECT RADIATION WTH GRAVITY AIR CIRCULATION.—*Technical Advisory Committee:* H. F. Hutzel, *Chairman;* R. M. Conner, R. E. Daly, R. V. Frost, J. H. Holton, A. P. Kratz, J. F. McIntire, W. T. Miller, R. N. Trane, O. G. Wendel.

The study of direct and indirect radiation was given consideration by the Technical Advisory Committee during the year and the investigation of the subject under the direction of Prof. A. C. Willard was made at the University of Illinois in cooperation with the Society. The subject has been broadened to include other phases related to direct and concealed radiation with gravity air circulation. In addition to the fully equipped room heating testing plant previously provided by the University, the latter has, during the past year, provided the recording and measuring instruments required for the operation of an eupatheoscope.

The investigation since January 1, 1932, has proceeded with three principal objectives: (1) The continuation of the study of the performance characteristics of different types of convectors when subjected to the actual service environment furnished by the room heating testing plant; (2) a study of the performance of convectors in the warm wall testing booth as provided for in the A. S. H. V. E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code), for the purpose of testing the validity of the correction factor used to convert the actual heat output under test conditions to the equivalent heat output under standard conditions (steam at 215 F and inlet air at 65 F); (3) a correlation of the performance of convectors in the warm wall booth with the performance in the room heating testing plant.

The published results of the entire investigation to date appear in three Bulletins, Nos. 169, 192 and 223, and one Reprint, No. 1, of the Engineering Experiment Station of the University of Illinois, and in six professional papers presented before the Society. The three papers appearing since January 1, 1932, are:

Performance of Convector Heaters, presented at the Annual Meeting of the Society, January 1932, and published in A. S. H. V. E. TRANSACTIONS, Vol. 38, 1932.

Tests of Convectors in a Warm Wall Testing Booth, presented at the Semi-Annual Meeting of the Society, June 1932, and published in A. S. H. V. E. TRANSACTIONS, Vol. 38, 1932.

Study of the Application of Thermocouples to the Measurement of Wall Surface Temperatures, presented at the Annual Meeting of the Society, January 1933, and published in A. S. H. V. E. TRANSACTIONS, Vol. 39, 1933.

7. ATMOSPHERIC DUST AND AIR CLEANING DEVICES.—*Technical Advisory Committee:* H. C. Murphy, *Chairman;* J. J. Aeberly, J. J. Bloomfield, Albert Buenger, Philip Drinker, Dr. E. V. Hill, H. B. Meller, Games Slayter, Perry West, Dr. S. W. Wynne.

The efforts of the Technical Advisory Committee during the past year have included laboratory investigations at the University of Minnesota and committee work which included the development of a code for testing air cleaning devices. The Technical Advisory Committee has cooperated with a national committee representing several other societies and associations interested in the dust problem. The general aim of this combined national committee is to develop a comprehensive study of the entire dust and smoke abatement, to promote general recognition of the problem by the public, and further the organization of means for alleviating the smoke and dust nuisance.

In the work at the University of Minnesota which is a cooperative project sponsored by the Society and the *National Warm Air Heating Association*, it has been necessary first to do a considerable amount of experimental work to determine the practicability of a proposed test code. Some of the difficulties were to obtain uniform distribution of dust, to select an acceptable type of dust, and to agree upon a method

of determining the efficiency. In this work, the test apparatus has been set up, the provisions of a test code have been checked, and the code has now been submitted.

Active cooperation has also been enjoyed by the committee with the Pure Air Committee of the *American Society of Mechanical Engineers.* The aim of this cooperation has been to coordinate all smoke abatement efforts throughout the country. Further cooperation has been carried on between the committee and other organizations, including Government Bureaus, and the Research Hospital of the University of Illinois at Chicago, where the effect of air impurities on certain allergic disorders, such as seasonal hay fever and bronchial asthma, has been investigated. The report of the investigations of the Research Hospital for the year was published in the May 1932 *Journal of the American Medical Association.* The results of this year's investigations were reported in a paper presented before the meeting of the Chicago Allergy Society and will be published in the *Journal of the American Medical Association* at a future date by Dr. Welker.

8. VENTILATION OF GARAGES AND BUS TERMINALS.—*Technical Advisory Committee:* E. K. Campbell, *Chairman;* T. M. Dugan, E. C. Evans, L. A. Harding, F. H. Hecht, V. W. Hunter, F. C. McIntosh, H. Lee Moore, A. H. Sluss.

The study of garage ventilation and the elimination of carbon monoxide from such establishments has received the attention of the Technical Advisory Committee during the past two years. The needs of the problem were originally brought to the attention of the Laboratory through the development of a code for the heating and ventilation of garages by the *National Fire Protection Association* in 1929 in cooperation with other interested organizations, including the Society.

Complete plans for the study were outlined by the Committee early during 1932, which included (1) a study of the ventilation of an underground ramp garage in a 40-story office building in Pittsburgh; (2) a study of the ventilation of a small one-story single room garage in Pittsburgh; and (3) a study of the ventilation of a garage in Lawrence, Kansas, by Prof. A. H. Sluss at the University of Kansas in cooperation with the Research Laboratory. These investigations were carried out during the early part of the year, resulting in the papers: Carbon Monoxide Distribution in Relation to the Ventilation of a One-Floor Garage, and Carbon Monoxide Distribution in Relation to the Ventilation of an Underground Ramp Garage, published in A. S. H. V. E. TRANSACTIONS, Vol. 38, 1932. Both papers were presented at the Semi-Annual Meeting of the Society in June 1932.

9. AIR CONDITIONS AND THEIR RELATION TO LIVING COMFORT.—*Technical Advisory Committee:* C. P. Yaglou, *Chairman;* I. C. Baker, W. L. Fleisher, D. E. French, Dr. E. V. Hill, Dr. R. R. Sayers.

The Technical Advisory Committee has been active during the past year in carrying out plans for research previously developed. Laboratory work under this committee was conducted during the year both at the Harvard School of Public Health under the direction of Prof. C. P. Yaglou and in cooperation with the Society, and at the Research Laboratory in Pittsburgh.

Cooperative research with Harvard is now being concentrated on subjective, physiologic, and biochemical effects of artificially ionized air, with the object of establishing whether or not artificial ionization definitely improves the quality of air, the conditions under which ionization may prove physiologically desirable or harmful, and the precautions to be taken in the application of ionization to conditioning air.

Plans for future work include a study of the minimum air requirements under sedentary conditions and, if finances permit, under conditions of muscular exercise also. The fundamental principles of the problem have been under investigation for the past two years at Harvard, and it is now proposed to apply these principles to large scale experiments in ordinary rooms for the purpose of checking the validity of the fundamental data under actual living and working conditions.

A study of the effect which cold walls have on the desired air temperature for feeling of comfort of a person in a room was made at the Laboratory in Pittsburgh and resulted in a paper, Cold Walls and Their Relation to Feeling of Warmth, A. S. H. V. E. TRANSACTIONS, Vol. 39, 1933, presented at the Annual Meeting 1933 in Cincinnati. This investigation shows that the effect of cold inside walls resulting from normal building construction and normal outside air temperatures, while in some sense important, is not a tremendous factor, unless the wall is of poor construction and the outside temperature extremely low.

10. RE-STUDY OF COMFORT CHART AND COMFORT LINE.—*Technical Advisory Committee:* C. P. Yaglou, *Chairman;* W. H. Carrier, Dr. E. V. Hill, F. C. Houghten, J. H. Walker.

This is a new Technical Advisory Committee organized at the beginning of the year and charged with the specific task of reviewing all previous work by the Society's laboratory in the development of the Effective Temperature Scale and Comfort Chart. The committee did a considerable amount of work during the first half of the year, resulting in a complete and comprehensive summation of the entire series of investigations relating to the comfort zone and comfort line, which should accomplish a great deal in bringing about a uniform understanding of the entire subject on the part of the membership of the Society. This report, entitled, How to Use the Effective Temperature Index and Comfort Charts, was presented at the Semi-Annual Meeting of the Society last June, and was published in A. S. H. V. E. TRANSACTIONS, Vol. 38, 1932.

11. OIL BURNING DEVICES.—*Technical Advisory Committee:* H. F. Tapp, *Chairman;* P. E. Fansler, R. V. Frost, F. B. Howell, R. C. Morgan, L. E. Seeley.

The Technical Advisory Committee on Oil Burning Devices has been active throughout the year in following out the laboratory studies by Prof. L. E. Seeley of Yale University in cooperation with the Society. A paper entitled, Air Supply and Its Effect on Performance of Oil Burners and Heating Boilers, resulted from this activity. (See A. S. H. V. E. TRANSACTIONS, Vol. 39, 1933.) The Committee has also been active in formulating new plans for future laboratory work.

12. CORRELATING THERMAL RESEARCH.—*Technical Advisory Committee:* R. M. Conner, *Chairman;* D. S. Boyden, P. D. Close, J. C. Fitts, W. T. Jones, H. T. Richardson, Perry West.

The Technical Advisory Committee on Correlating Thermal Research was active during the past year in the perfecting of the organization and promotion of the work under the auspices of the Research Laboratory. Formerly this work was carried on by a separate association which has now disbanded. A directory was prepared and distributed during the year, and the work of cataloging the available literature was continued.

13. REFRIGERATION IN RELATION TO AIR TREATMENT.—*Technical Advisory Committee:* A. P. Kratz, *Chairman;* E. A. Brandt, G. B. Bright, S. R. Lewis, H. J. Macintire, E. D. Milener, F. G. Sedgwick, A. R. Stevenson, Jr., J. H. Walker, R. W. Waterfill, A. C. Willard, A. W. Williams, H. M. Williams.

This is a new project undertaken by the Committee on Research during the past year. The Technical Advisory Committee was organized in the Spring. A Sub-Committee of the Technical Advisory Committee met in Columbus, Ohio, in May and outlined a program of research to be carried on at the Research Residence of the University of Illinois in cooperation with the Society. This program was later approved by the entire Technical Advisory Committee. A paper entitled, Study of Summer Cooling in the Research Residence at the University of Illinois, by A. P. Kratz and S. Konzo, resulted from this investigation. (See A. S. H. V. E. TRANSACTIONS, Vol. 39, 1933.)

A coordinated study of the same subject was also made during the summer by J. H. Walker and G. B. Helmrich of the Detroit Edison Company in cooperation with the Laboratory. A paper entitled, Summer Cooling Operating Results in a Detroit Residence, resulted from this work. (See A. S. H. V. E. TRANSACTIONS, Vol. 39, 1933.)

For the purpose of this investigation, the *National Warm Air Heating Association* also cooperated by providing the use of the Research Residence and other services.

14. HEAT TRANSMISSION.—*Technical Advisory Committee:* L. A. Harding, *Chairman;* A. B. Algren, R. E. Backstrom, A. H. Barker, A. E. Stacey, Jr., J. H. Walker.

The subject of heat transmission has been under investigation since the organization of the Laboratory and a great number of papers on this subject have been published. Based on these papers, the section on heat loss from buildings in THE A. S. H. V. E. GUIDE has been greatly improved and developed. These investigations have included heat flow measurements in actual buildings made with the Nicholls' Heat Flow Meter developed at the Laboratory, effect of aging on the conductivity of concrete, and the effect of heat capacity and solar radiation on heat flow in a building, at the Laboratory. A study of film conductance coefficients for

various surfaces in still and moving air was made at both the Laboratory and the University of Minnesota. Studies were also made of heat flow through built-up wall panels and insulating and building materials, and of the insulating effect of air spaces at the University of Minnesota in cooperation with the Laboratory.

Financial Report of Research Laboratory of the American Society of Heating and Ventilating Engineers—Pittsburgh, Pa.

For the Year Ended December 31, 1932

CASH RECEIPTS AND DISBURSEMENTS

BALANCE—JANUARY 1, 1932—PER FORMER REPORT			$ 7,049.58
RECEIPTS			
From the American Society of Heating and Ventilating Engineers	$16,795.92		
Contributions—Per Schedule	10,765.00		
Cleveland Exposition	5,012.76	$32,573.68	
Interest on Bank Balances	36.41		
Interest on Securities—Research	135.00		
Interest on Securities—Research Endowment Fund	1,025.00	1,196.41	33,770.09
			40,819.67
DISBURSEMENTS			
Salaries—Per Schedule		12,449.16	
Technical Adviser—Committee on Research		2,800.00	
Traveling—Executive Committee	556.99		
Traveling—F. C. Houghton	231.99		
Traveling—Staff	98.77	887.75	
Clerical—New York		800.00	
Laboratory Equipment and Supplies		571.71	
Correlating Thermal Research		500.00	
University Cooperation—Per Schedule		12,789.50	
Cleveland Exposition Exhibit		432.34	
Meetings		67.25	
OFFICE EXPENSES			
Stenographic Services (Committee)	162.25		
Postage	125.83		
Stationery and Printing	275.90		
Telephone and Telegraph	42.01		
Professional Services	100.00		
Expressage	66.60		
Miscellaneous	132.25	904.84	32,202.55
BALANCE—DECEMBER 31, 1932			$ 8,617.12

BUDGET COMPARISON

INCOME	ACTUAL	BUDGET PROVISION	INCREASES DECREASES
Dues—From the American Society of Heating and Ventilating Engineers	$16,795.92	$20,000.00	$ 3,204.08
Manufacturers	4,740.00	5,000.00	260.00
Associations	3,525.00	3,000.00	525.00
Exposition (Net)	5,012.76	2,250.00	2,762.76
Other Contributions	2,500.00	5,000.00	2,500.00
Interest on Bank Balances	36.41	60.00	23.59
Interest on Endowment Fund	1,025.00	1,000.00	25.00
Interest on Securities	135.00	100.00	35.00
Surplus Fund	—0—	1,750.00	1,750.00
	$33,770.09	$38,160.00	$ 4,389.91

EXPENSES			
COMMITTEE ON RESEARCH			
Travel—Committee and Laboratory Personnel..	$ 887.75	$ 2,000.00	$ 1,112.25
Salary—Clerical, Society Headquarters, 1931..	800.00	800.00	—0—
Salary—Technical Adviser..................	2,800.00	2,800.00	—0—
General Printing and Promotion............	212.25	500.00	287.75
Correlating Thermal Research...............	500.00	500.00	—0—
	$5,200.00	$6,600.00	$ 1,400.00
LABORATORY (PITTSBURGH)			
Salaries..................................	12,449.16	15,000.00	2,550.84
Laboratory Supplies and Equipment.........	571.71	1,500.00	928.29
Office Supplies and Expense................	692.59	750.00	57.41
Meetings...................................	67.25	200.00	132.75
	$13,780.71	$17,450.00	$ 3,669.29
CONTRACTS WITH COOPERATING INSTITUTIONS			
Contracts..................................	12,789.50	13,175.00	385.50
Emergency Fund..........................	—0—	935.00	935.00
EXPOSITION EXHIBIT....................	432.34	—0—	432.34
	13,221.84	14,110.00	888.16
	32,202.55	38,160.00	$ 5,957.45
	$ 1,567.54	$ —0—	$ 1,567.54

Following the report of the Committee on Research, Dr. C. A. Mills,[1] Cincinnati, O., offered some interesting comments on the relationship between the work of the physiologist, the physician, and the engineer.

C. A. MILLS, M.D.: I am not a member of your Society, nor in any way a heating or ventilating engineer, so that I feel hesitant in addressing your convention. As a physiologist and physician, however, I am very much interested in the results of your work as it affects the health and welfare of man. The great bulk of your efforts are expended on buildings designed to house man, so that it becomes a very great problem in public and individual health to determine more exactly just what air conditions are best for man under different situations. The southerner, for instance really needs a higher indoor temperature in winter than do northerners, because his own internal heat production is more sluggish and he chills more easily.

For the past 5 years I have been studying on man and animals the effects of varying environmental conditions. These studies have given strong indications that our atmospheric environment in large measure governs the energy level of our existence, our vigor and vitality, as well as our fertility of mind and body. As a result of these studies I should like to stress three points for your consideration in relation to your development of the field of air conditioning.

First, we can now specify conditions which, if applied to people of the tropics and sub-tropics (or our Gulf region), would raise their energy level up close to that of the energetic northerner, doing away in large measure with the traditional southern physical inertia. It only involves a few hours cooling each day, which could be applied at night in sleeping quarters. Such cooling would so increase the energy and efficiency of mankind as to make it a profitable change for all concerned. It would also increase the nervous tension and restlessness of the people, probably reducing their total pleasure in life and making them more susceptible to

[1] Professor of Experimental Medicine, University of Cincinnati.

such diseases as diabetes and high blood pressure. To live faster and on a higher economic level seems to be the predominant American aim, however, and this would be promoted by cooling, properly applied, in the South. There would, however, be a lag of several years before effects would be very apparent, so that the introduction of such air conditioning on a large scale will be very difficult unless it can be accomplished at low cost.

My second point concerns air conditioning in hospitals and I shall stress only one item. In our severe summer heat waves we are afflicted with epidemics of acute appendicitis, attacks that fulminate rapidly, often going into peritonitis within a day or two, and demanding immediate operation. These attacks seem related to the heat waves, and must at present be operated and kept in hot hospital rooms, although high temperatures add greatly to the hazards for the patients. Appendicitis cases form a large part of every hospital's population, particularly in the smaller hospitals. It would be of exceedingly great value if we could have complete air conditioning during these summer heat waves, saving many lives and speeding recovery of those afflicted. I have mentioned only acute appendicitis, but the application of cooling would be equally valuable for children with summer diarrhea which we now know is closely dependent on these heat waves. Victims of heat exhaustion would also benefit greatly, and I believe by proper variation in completely controlled air we could shorten the convalescence from many operations, fevers, and obstetrical procedures.

It is my belief that we are going to build few large hospitals in the coming decade, and air conditioning those we have is an expensive procedure. However, it is likely that there will soon develop a movement for the erection of countless small community health centers throughout the country. There will not be much money available for these buildings, so that, if you can lower construction and upkeep costs sufficiently to allow the installation of air conditioning and sound deadening you will have performed a major service to humanity.

The third point concerns ventilation. The cost of air conditioning is predicated on the turn-over of air required, and in past years we have been fresh air fiends. I believe, however, that there is beginning a strong shift in the opposite direction, and that we will pay more attention to the state of the air than to its rapid replacement. We never seriously reduce the oxygen of closed rooms to vitiate the air, nor does the carbon dioxide rise to dangerous heights. Most recent studies indicate that the factor affecting us most is the ionic state. An article setting forth facts along this line is one by Lewis R. Koller.[2] The application of these facts to heating and ventilating has to do largely with the effects of overheating the air, particularly in contact with hot metals. The deadening of air so often encountered in warm air systems is due to this overheating as the air comes in contact with the hot furnace metal. In general the cooler we can keep the air we breathe the more vitalizing it will remain. Therefore I believe it will pay you to give particular attention to construction methods which will reduce the amount of heating needed and then to apply this heat by low-temperature radiation as far as possible.

The paper, Some Observations on Heating Practice, presented this morning by James Govan interested me intensely. He has, I believe, solved the construction and heating question to give us buildings in which the building and maintenance costs can be sufficiently reduced to permit complete air conditioning and sound control. With the present economic resistance to high building costs, Mr. Govan's program would seem to be particularly timely. Air conditioning in the South and throughout the tropics must needs be supplied at extremely low cost because of the great lag in its financial return to the people there. In the smaller hospitals that will soon go up in great numbers, on the other hand, you would see immediate benefits from such control of indoor environment. I believe nothing would do more

[2] *Jour. of the Franklin Institute*, 1932, Vol. 214, p. 543.

to aid the eventual application of complete air conditioning in homes all over the country than its successful and economical use in hospitals.

As I said in the beginning I am greatly interested in the human side of your work. I can only urge you to lend every effort toward making more available control of our indoor environment, and bringing it within the reach of every one.

At the second session, Monday, January 23, 2:00 p. m., President Rowley called for the report of the Tellers of Election, which was given by A. W. Williams, Chairman.

Report of Tellers

Your Board of Tellers wishes to report results of the vote for Officers and Council and members of the Committee on Research:

President—W. T. JONES	430
First Vice-President—C. V. HAYNES	424
Second Vice-President—JOHN HOWATT	431
Treasurer—D. S. BOYDEN	432
Members of the Council—Three Year Term:	
R. H. CARPENTER	430
J. D. CASSELL	428
F. C. MCINTOSH	429
L. W. MOON	430
Members of the Committee on Research—Three Year Term:	
ALBERT BUENGER	432
S. H. DOWNS	432
H. N. KITCHELL	432
H. R. LINN	431
PERRY WEST	432
One Year Term:	
E. N. SANBERN	432
Total ballots cast	480
Disqualified	48

TELLERS OF ELECTION,
A. W. WILLIAMS, *Chairman*
R. J. TENKONOHY
A. W. ROOKS

Report of Committee on Revision of Constitution and By-Laws

President Rowley called upon W. T. Jones, Chairman of the Committee on Revision of Constitution and By-Laws. Mr. Jones explained the set-up of the new Constitution, By-Laws, and Rules and how it differed from the present Constitution and By-Laws. He pointed out that each member had received a copy and said that some valuable suggestions had been received. At the conclusion of Mr. Jones' analysis, a motion was made by D. S. Boyden, seconded by R. H. Carpenter, as follows:

Resolved, that the report of the Committee on Revision of Constitution and By-Laws be adopted.

Following a discussion, the members present then voted on a motion to adopt the new Constitution as prepared by the Committee. The vote was overwhelmingly in favor.

F. D. Mensing presented a resolution as follows and moved its adoption:

Resolved, that the new By-Laws be accepted.

The motion was seconded by J. D. Cassell.

An amendment was offered to Article B-XI by John Howatt, Chicago. He proposed the addition of *Section 8,* as follows:

A Society Reserve Fund shall be created into which all admission fees and such other monies as the Council may direct shall be placed until the fund totals a sum equal to $15.00 per member. This Reserve Fund is to be used only in cases of emergency when current Society revenues are insufficient to pay necessary expenses. Withdrawals not to exceed 20% of the fund in any calendar year may be authorized by the Society at any meeting provided it has been recommended by the Council. Investments shall be made in accordance with *Section I* of Article B-XI. Interest earned on the Reserve Fund shall be added to current Society income.

His motion to adopt was seconded by J. D. Cassell, and was unanimously approved by vote of the members.

F. D. Mensing proposed the following revision of *Section 1,* Article B-XI, relating to funds:

When so directed by Council, the Chairman of the Finance Committee shall invest such portion of any funds of the Society as determined by the Council in securities of the United States Government. All such investment shall be approved by Council.

This was seconded by J. D. Cassell, and when presented for vote, was adopted unanimously.

In order that other Articles might conform, F. D. Mensing proposed the elimination from *Section 3,* Article B-XI, of the words, "legal for the investment of funds of savings banks of the State of New York provided by *Sections 1* and *2.*"

This motion was seconded by Homer Linn, Chicago, and approved.

W. T. Jones inquired if there were further amendments and then presented the motion to adopt the By-Laws as amended with the exception of Article B-IV and *Section 10* of B-VIII. The motion was carried.

A motion was proposed by W. R. Eichberg, of Philadelphia, as follows:

Resolved, that the Rules be adopted as written by the Committee.

The motion was seconded by J. D. Cassell and the vote of the members was in favor of adoption.

F. D. Mensing, Philadelphia, proposed the following motion:

Resolved, that Appendix "A" of the Constitution, By-Laws, and Rules which are the Regulations Governing the Committee on Research, be adopted as written.

The motion was seconded by W. R. Eichberg and carried.

F. D. Mensing also proposed the following motion:

Resolved, that Appendix "B," the Rules Governing the Award of the F. Paul Anderson Medal, be adopted as written.

This motion was seconded by J. D. Cassell and unanimously carried.

Chairman W. T. Jones announced that suggestions regarding dues and the style of the Nominating Committee were being prepared by the Advisory Council.

At a later session, Mr. Jones took the chair and explained the idea of the Committee on Constitution and By-Laws regarding the make-up of the Nominating Committee and stated that when the plan was presented at the Council not all were in agreement. It was, therefore, decided to present this matter with the question of dues rate to the Advisory Council.

W. H. Driscoll, expressing the views of the Advisory Council, suggested

the matter of admission fees and dues be discussed first. He then presented the following motion, that *Section 1,* Article B-IV, read as follows:

That the admission fee of members, associate members, junior members and student members shall be as determined by the Council until 1934, and thereafter the admission fee of members, and associate members, shall be $15.00, of junior members $5.00, and of student members, $2.00. Admission fee must accompany application.

Further—that *Section 2* read as follows:

The annual dues of members, associate members, junior members and student members shall be as determined by the Council until 1934, and thereafter the annual dues of members, and associate members shall be $25.00, of junior members $12.00, and of student members, $3.00.

The motion was seconded by C. V. Haynes and on vote was adopted.

On motion of W. H. Driscoll, seconded by C. V. Haynes, it was voted that *Sections 3, 4, 5, 6, 7, 8, 9* and *10* of Article B-XI of the By-Laws be adopted as written. This motion was passed.

W. H. Driscoll then presented the ideas of the Advisory Council regarding the Nominating Committee, which coincided with the views of the Committee on the Revision of Constitution and By-Laws, with the exception of the addition of the following sentence:

No proxies shall be permitted.

This matter was discussed in considerable detail by Messrs. Lyle, Evans, Pickett, and a number of others, with the result that the proposed *Section 10* was amended to read as follows:

The Nominating Committee shall consist of one member eligible to vote designated by each chapter or his alternate also appointed by the chapter. The secretary of each chapter shall certify to the secretary of the Society on or before January 1, the names of the member and alternate selected.

The Committee shall meet at the Annual Meeting of the Society at the call of the Secretary of the Society and shall effect its own organization and elect its own Chairman. At the Semi-Annual Meeting of the Society, if possible, the Nominating Committee shall select the nominees for the ensuing year for the offices of President, First Vice-President, Second Vice-President, Treasurer, and four (4) members of the Council. In any event the names of the nominees shall be certified to the Secretary of the Society before September 20, with the written consent of each nominee to fill the office for which he has been selected and their names with the offices to which they have been nominated shall be published in the October issue of the JOURNAL.

The motion was seconded by W. R. Eichberg and passed on vote of the members present.

A motion was then offered to have the Constitution, By-Laws and Rules as amended, submitted to the entire membership for letter ballot. This motion was carried.

The Constitution and By-Laws as adopted appear on page 27.

President Rowley called the third session to order on Tuesday, January 24, at 9:30 a. m., and introduced D. S. Boyden, Chairman of the Guide Publication Committee, who gave his report.

Report of Guide Publication Committee

Of all the branches of the building industry, perhaps none has made greater strides and has had more changes with respect to equipment, practices and design pro-

cedures than heating and ventilation, and its special phase—air conditioning. Because of this fact, the A. S. H. V. E. GUIDE has been unique in the building field in that it has kept pace with these rapid changes and developments by being revised and brought up to date annually. It has thereby provided the industry with the latest available design data and information resulting from the research investigations of the Society and the practices of the outstanding authorities in this field.

The present issue, THE GUIDE 1933, is no exception to this general policy of keeping THE GUIDE up to date. In fact, more changes and new material are included in this issue than in any previous issue. Furthermore, THE GUIDE 1933 contains 50 more pages of text matter and five more chapters than the next largest issue, its predecessor, THE GUIDE 1932.

The Text Section includes 11 entirely new chapters, namely, Chapter 7, Radiant Heating; Chapter 12, Domestic Water Requirements; Chapter 21, Thermodynamics of Air Conditioning; Chapter 23, Mechanical Warm Air Heating and Air Conditioning Systems; Chapter 25, Central Fan Cooling and Air Conditioning Systems; Chapter 26, Cooling and Dehumidifying; Chapter 27, Refrigeration as Applied to Air Conditioning; Chapter 31, Control of Air Conditioning Systems; Chapter 32, Air Distribution within Enclosures; Chapter 34, Sound Control; and Chapter 37, Exhaust Systems.

In addition to the many new subjects introduced in these chapters, others have been extensively or completely revised, and practically none of the old copy was retained. Chapter 3, dealing with Transmission Losses, is almost entirely new and contains a completely revised set of tables of coefficients of transmission based on the latest researches of the Society at the University of Minnesota, and at the Laboratory in Pittsburgh. A large portion of the material in Chapter 14, Heating boilers, has been replaced by more up-to-date information, and the chapter has been enlarged. One of the most important chapters of the present issue of THE GUIDE is Chapter 22 dealing with Ventilation and Air Conditioning Standards for Comfort and Health, which has also been revised in the light of results obtained from recent researches at Harvard University and at the Laboratory of the Society in Pittsburgh. This chapter also includes the ventilation standards adopted by the Society in August, 1932.

Central Fan Heating and Air Conditioning Systems are treated in Chapter 24. This chapter contains a considerable amount of information and data which have not heretofore been published, whereas many changes were made in the material on the design of air ducts (Chapter 33) including the introduction of new and more complete examples showing the latest methods of arriving at accurate sizes.

In Chapter 36, which deals with Natural Ventilation, a completely new treatise of this subject has been given. Extensive changes have also been made in Chapter 38, dealing with Fans and Motive Power, whereas the second part of Chapter 29, dealing with Test Instruments and Methods, is entirely new.

Other chapters in which numerous revisions were made include Chapter 4, Infiltration Heat Losses; Chapter 6, Radiators and Gravity Convectors; Chapters 9 and 10 dealing with Steam Heating Systems; Chapters 16, 17 and 18 dealing with fuels and appliances for the combustion thereof; and Chapter 43 on Smoke and Dust Abatement.

The 45 chapters and 592 pages of text matter in THE GUIDE are supplemented by a valuable Catalog Data Section of Modern Equipment in which detailed descriptions, sizes, capacities, and dimensions of various commodities and appurtenances are given.

GUIDE PUBLICATION COMMITTEE
D. S. BOYDEN, *Chairman*

W. L. McGrath, President of the *National Warm Air Heating Association,* was introduced and presented greetings from the Association and expressed his satisfaction in the results of the cooperative work being done at the University of Illinois.

A. Bachman, Secretary of the *Cincinnati Association of Heating and Piping Contractors,* presented greetings from the National Association and told of the necessity for cooperative effort in making progress in our field.

Appreciation of the work of the Society was expressed by Patrick Sullivan, representative of the *United Association of Journeymen, Plumbers and Steamfitters.*

The fourth session on Tuesday, January 24, at 2:00 p. m. was opened by President Rowley, who introduced E. K. Campbell, Chairman of the Chapter Relations Committee.

Report of Chapter Relations Committee

The activity of the Chapter Relations Committee during the past year has been divided between two lines of effort. First, to help those Chapters desiring it, by providing speakers and offering other program suggestions and second by promoting joint meetings between the Chapters.

In providing speakers we have tried to bear in mind that these are hard times. The Chapters have most of them reduced their dues and haven't much money in the treasury for the payment of speakers' expenses. So we have tried to get speakers who while adhering strictly to the standards of the Society regarding advertising, yet in one way or another would feel repaid so they could pay their own expenses.

These speakers have according to the reports, given uniformly desirable and happy talks which have pleased the Chapters who have heard them very much.

During the year 28 meetings have been arranged by the Chapter Relations Committee either by providing a speaker or by suggesting a speaker who has been arranged for directly by the Chapters or by suggestion of motion pictures or other features.

It is unfortunate that the Committee has been of very little assistance to the Pacific Coast Chapters and it is the more to be regretted because contact between those Chapters and the parent body is more difficult because of the distance involved.

It is evident that the larger Chapters have little need for the services of the Committee because they have such a wealth of material available for their meetings at all times, and right at their doors. They, of course, have more intimate knowledge of those possibilities than the committee could have.

We ask the Chapters in the larger cities to help the committee by passing on to us information about speakers who might be available for other Chapters and whose talks have been pretty good. We ask them to pass on any ideas they may have to the Committee other Chapters might be interested in.

We do not regard the Chapter Relations Committee as a source of ideas but rather as a clearing house for ideas and we hope that as the Chapters find things that are of particular interest and value, that they will pass them on to the committee.

It is to be hoped that as the results of the Chapter Relations Committee become better known and understood, that the committee may be of more service to all of the Chapters and particularly those who have the need for it.

The committee has had an idea that it would be well to promote cordial relations between the Chapters as well as to help in the Chapter meetings, and with that in view it persuaded the Kansas City Chapter to extend an invitation to the St. Louis Chapter for a joint meeting to be held in Kansas City. To this meeting came 10 of the St. Louis Chapter bringing with them Professor Willard as the speaker of the meeting. This resulted in a considerable extension of acquaintanceship between the membership of the two Chapters and the meeting was so successful that before it was over, a spontaneous demand arose for another joint meeting which was actually held in Urbana at the Research Laboratory of the University of Illinois, Oct. 15th. To this meeting were invited the membership at large and particularly the members of Illinois and Wisconsin Chapters in addition to those of St. Louis and Kansas City. All four Chapters had representatives there, with a total attendance of about 35.

It is understood that the Minnesota and Wisconsin Chapters have a joint meeting schedule to be held some time in the spring at some half way point, probably La Crosse.

It seemed to the Chapter Relations Committee that joint meetings of this sort even though not attended by a large percentage of the membership, would promote

acquaintanceship between the Chapters in a very happy way and the few joint meetings that have been held have proved out just that way and we strongly urge that more Chapters try out the joint meeting idea and we offer the services of the Committee toward arranging such meetings, wherever we can be of service.

CHAPTER RELATIONS COMMITTEE
E. K. CAMPBELL, *Chairman*

At the closing session on Wednesday, January 25, at 9:30 a. m., the installation of officers was conducted by Messrs. L. A. Harding, G. L. Larson and E. K. Campbell. The new president, W. T. Jones, Boston, was escorted to the platform and introduced. The gavel was turned over to him by the retiring president, Professor Rowley. First Vice-President C. V. Haynes, Philadelphia, and Second Vice-President John Howatt, Chicago, were presented to the members and the Treasurer, D. S. Boyden, Boston, was introduced. The members of the Council to serve for three years were escorted to the platform and presented to the members.

Resolutions

With President Jones presiding, several resolutions were offered.

W. H. Driscoll presented the following motion:

Resolved, that the Secretary of the Society be instructed to communicate with our fellow member, Mr. Andrew C. Edgar, of Philadelphia, and express to him the hope of the members assembled that his recovery to health will be speedy and to assure him that his record with the Society of having attended each of its 39 Annual Meetings remains intact and, further, inform him that the spirit of his presence was felt at the Annual Banquet even though he could not occupy the seat reserved for him.

The motion was seconded by J. J. Aeberly and unanimously adopted.

The Report of the Resolutions Committee was presented by the chairman, M. F. Blankin, of Philadelphia, and the following resolutions were unanimously adopted by rising vote:

Be It Resolved, that the Cincinnati Chapter, A. S. H. V. E., in spite of the handicap of being less than six months old has rendered most distinguished service in conducting this Thirty-ninth Annual Meeting and that the A. S. H. V. E. here assembled do hereby express their most hearty and sincere thanks to the Chapter and the hard working committee.

Be It Further Resolved, that Mr. Sam Fowlkes, the Convention Manager of the Gibson Hotel, has been most gracious and efficient in giving us unparalleled service, and that we herewith tender our keen appreciation to him, the staff and the management.

Be It Further Resolved, that Cincinnati's Convention Committee have rendered most valuable cooperation and service in connection with our Thirty-ninth Annual Meeting and that we hereby extend to them our hearty commendation.

Be It Further Resolved, that the Cincinnati newspapers have given us a far greater amount of publicity than ever before received in our thirty-nine years of existence and that we do wish to express our most sincere gratitude.

Be It Further Resolved, that the Society's thanks be extended to the trade press for their attendance at this Convention in greater numbers than ever.

Be It Further Resolved, that all the railroads coming into Cincinnati have given us most courteous cooperation, especially the Baltimore and Ohio Railroad in connection with providing transportation and unusual accommodations in sending home our beloved member, Mr. A. C. Edgar, who was taken sick while attending the Meeting.

Be It Further Resolved, that a copy of these resolutions be spread upon the minutes of this meeting and the Secretary be instructed to send letters of appreciation to each of the party or parties referred to advising them of the action of the Convention.

At the call for new business W. H. Driscoll proposed the following resolution which was seconded by W. W. Timmis:

Resolved, that a committee be appointed by the president to consider the question of the elimination of the term *"square foot,"* now used in connection with the ratings of heating boilers and radiators, and the substitution therefore of a new term or expression that will properly and adequately indicate a unit of heat transmission, not only as applied to the ratings of boilers and radiators, but sufficiently comprehensive to be applicable to related arts and industries: such as warm air heating, unit heating, refrigeration, fuels, etc., *and Be It Further Resolved,* that this committee be instructed to confer with the *Institute of Boiler and Radiator Manufacturers,* the *Heating and Piping Contractors National Association,* the *American Society of Refrigerating Engineers,* and other interested trade associations and engineering societies in order to obtain their advice, assistance and cooperation; *and Further,* that the committee be instructed to make a report and recommendation to the Society at as early a date as possible.

The resolution was adopted.

As no more new business was offered, announcement was made of a Council meeting immediately following the session and the 39th Annual Meeting of the Society adjourned.

PROGRAM 39TH ANNUAL MEETING

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
HOTEL GIBSON, CINCINNATI, OHIO
JANUARY 23-25, 1933

Sunday, January 22

10:00 A.M. Meeting of the Council (Gold & Ivory Room)
2:00 P.M. Meeting of Committee on Research (Adams Room)
5:00 P.M. Meeting of Committee on Ventilation Standards

Monday, January 23
(Roof Garden)

9:30 A.M. Introduction—K. A. Wright, President Cincinnati Chapter
Greeting—Hon. Russell Wilson, Mayor of Cincinnati
Response—President F. B. Rowley
Reports of Officers
Report of Council
Reports of Committees—Finance, Membership, Publication
Technical Papers:
Study of the Application of Thermocouples to the Measurement of Wall Surface Temperatures by A. P. Kratz and E. L. Broderick
Some Observations on Heating Practice by James Govan
Report of Committee on Research—Prof. G. L. Larson
Report of Tellers—A. W. Williams, *Chairman*
12:30 P.M. Ladies' Luncheon—Bird of Paradise Room, Hotel Gibson
2:00 P.M. Bridge Tea for Ladies—Golden Palm Room, Hotel Gibson
2:00 P.M. *Technical Paper:*
Air Supply and Its Effect on Performance of Oil Burners and Heating Boilers by L. E. Seeley, J. H. Powers and E. J. Tavanlar
Report of Committee on Revision of Constitution and By-Laws by W. T. Jones
6:30 P.M. Dinner for Past Presidents (Gold and Ivory Room)
7:30 P.M. Discussion of Report of Committee on Revision of Constitution and By-Laws (Club Room—10th Floor)
8:30 P.M. A Night in Old Cincinnati (Entertainment, Dance and Buffet Supper)—Roof Garden, Hotel Gibson

Tuesday, January 24
(Roof Garden)

9:00 A.M. Meeting of the Nominating Committee (Gold and Ivory Room)
9:30 A.M. Report of Guide Publication Committee—D. S. Boyden

Technical Papers:
Cold Walls and Their Relation to the Feeling of Warmth by F. C. Houghten and Paul McDermott
Study of Summer Cooling in the Research Residence at the University of Illinois by A. P. Kratz and S. Konzo
Summer Cooling Operating Results in a Detroit Residence by J. H. Walker and G. B. Helmrich

10:00 A.M. Ladies' Motor Trip Seeing Cincinnati—(Taft Museum and Rookwood Pottery)
12:30 P.M. Ladies' Luncheon—Florentine Room, Hotel Gibson
2:00 P.M. Ladies' Theatre Party
2:00 P.M. Report of Committee on Chapter Relations—E. K. Campbell

Technical Papers:
Corrosion in Air Conditioning Equipment and Its Prevention by R. M. Palmer
Air Supply, Distribution and Exhaust Systems by S. R. Lewis
Cow-Barn Ventilation by A. J. Offner
Air Infiltration through Steel Framed Windows by D. O. Rusk, V. H. Cherry and L. Boelter

7:00 P.M. Annual Banquet and Dance—Roof Garden, Hotel Gibson

Wednesday, January 25
(Roof Garden)

9:30 A.M. *Technical Papers:*
Flow of Condensate and Air in Steam Heating Returns by F. C. Houghten and Carl Gutberlet
Condensate and Air Return in Steam Heating Systems by F. C. Houghten and J. L. Blackshaw

Installation of Officers
New Business
Resolutions

12:30 P.M. Luncheon Meeting—Council

COMMITTEE ON ARRANGEMENTS

W. C. GREEN, *General Chairman*

H. N. KITCHELL, *Vice-Chairman*

Publicity: J. J. Braun, *Chairman;* K. A. Wright, F. D. Mensing.
Ladies: R. B. Breneman, *Chairman;* I. B. Helburn, H. N. Kitchell.
Entertainment: I. B. Helburn, *Chairman;* R. B. Breneman, H. N. Kitchell.
Finance: C. J. Kiefer, *Chairman;* W. J. Doyle, R. E. Peck.
Reception: A. W. Rooks, *Chairman;* R. W. Sigmund.
Registration: E. B. Royer, *Chairman;* O. W. Motz, R. W. Sigmund.
Banquet: H. E. Sproull, *Chairman;* O. W. Motz, H. A. Pillen, A. Winther.
Transportation: C. E. Hust, *Chairman.*

No. 942

CONSTITUTION AND BY-LAWS*

of the

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

CONSTITUTION

ARTICLE C-I—Name, Objects and Government

Section 1. The name of this Society is the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Section 2. The Society is a corporation organized September 10, 1894, and chartered under the laws of the State of New York, January 24, 1895.

Section 3. The Society is organized to advance the arts and sciences of heating, ventilating and air conditioning by research, discussion, intercourse and publication.

Section 4. The Society shall be governed by this Constitution, the By-Laws and the Rules.

ARTICLE C-II—Membership

Section 1. Persons connected with the arts and sciences related to heating, ventilating or air conditioning are eligible for admission into the Society.

Section 2. The membership of the Society shall consist of Honorary Members, Members, Junior Members, Associate Members and Student Members.

Section 3. An Honorary Member shall be a person of acknowledged professional eminence.

Section 4. A Member shall be thirty (30) years of age or over and shall be a person of experience in the science of heat transfer in its application to the art of heating, ventilating or air conditioning, and shall have been in active practice of his profession and in responsible charge of important work for five (5) years and shall be qualified to design as well as to direct such engineering work. Fulfilling the duties of a professor in one of the allied sciences in a college or technical school of accepted standing shall be taken as an equivalent to an equal number of years of active practice. Graduation from a school of engineering of recognized standing shall be considered as equivalent to two (2) years of active practice. (*Also see Section 9*).

Section 5. A Junior Member shall be a person over twenty (20) years and under thirty (30) years of age, who has been actively engaged in the work of heating, ventilating or air conditioning for three (3) years, or is a graduate of a school of engineering of recognized standing.

Section 6. An Associate Member shall be twenty-five (25) years of age or over. He need not be an engineer, but must have been so connected with some branch of engineering or the art of heating, ventilating, air conditioning or the industries relating thereto, that he may be considered as qualified to co-operate with heating and ventilating engineers in the advancement of professional knowledge.

*Adopted January 1933 by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Section 7. A Student Member shall be a person over sixteen (16) years of age who is regularly attending courses in an engineering college or school.

Section 8. All grades of membership shall have equal standing in the Society, excepting as follows:

(*a*) Junior Members and Student Members shall have no vote nor hold office in the Society.

(*b*) Associate Members shall be entitled to vote on all matters submitted, but shall not be entitled to hold office.

Section 9. Mining, civil, electrical, mechanical, naval or government engineers, chemists, physicians, scientists, or architects, who are qualified by reason of their experience in designing, improving, inspecting, investigating or developing the arts or sciences of heating, ventilating or air conditioning, are also eligible to membership.

ARTICLE C-III—Admission and Advancement

Section 1. Honorary Members shall be nominated by at least ten (10) members of the Society. The grounds upon which the nomination is made shall be presented to the Council in writing and be signed by the ten (10) proposers.

Section 2. Election to any grade of membership in the Society, except Honorary Membership shall be by an affirmative vote of two-thirds of the Council.

Section 3. When a Junior Member reaches the age limit of his grade, he shall be automatically transferred to Associate grade of membership, as provided in the By-Laws, unless he has applied for and has been elected to Member grade.

Section 4. When a Student Member discontinues his regular studies in an engineering college or school, it shall be incumbent upon him to apply within one (1) year for advancement to either Junior, Associate or Member grade.

ARTICLE C-IV—Admission Fees and Dues

Section 1. Admission fees and dues shall be as provided in the By-Laws.

ARTICLE C-V—The Council

Section 1. The affairs of the Society shall be managed by a Board of Directors chosen from among the persons entitled to vote, which shall be styled "The Council." It shall consist of the President, First Vice-President, Second Vice-President, Treasurer, the last Past President and twelve (12) elected members, four (4) of whom shall be elected each year to hold office for three (3) years. The Secretary may take part in the deliberation of the Council, but shall have no vote therein.

Section 2. The Council shall regulate its own proceedings and may delegate specific powers to any Committee or to any one or more members by resolution.

Section 3. Should a vacancy occur in the Council, or any elective office, through death, resignation or other cause, the Council may elect a Member to fill the vacancy, pending the next annual election.

Section 4. The members of the Council, or Directors will be held harmless by the Society from any liability to third persons, resulting from any acts in their capacity as such Directors or when engaged in the affairs or business of the Society. Any such liability caused by the wilful misconduct of any Officer will not be assumed by the Society.

Section 5. No act of any Committee or any delegate shall be binding until it has been approved by a resolution of the Council.

Section 6. An act of the Council, which shall have received the expressed or implied sanction of the membership at the next subsequent meeting of the Society, shall be deemed to be an act of the Society, and shall not afterwards be impeached by any member.

ARTICLE C-VI—Officers

Section 1. The Officers of the Society are the members of the Council.

Section 2. The President, Secretary and Treasurer shall perform the duties usually pertaining to their respective offices and such other duties as may be provided for in the By-Laws and Rules or required of them by the Council.

ARTICLE C-VII—Meetings

Section 1. The Society shall hold an Annual Meeting during the week beginning with the fourth Monday in January at such place as the Council may elect and continuing from day to day as the Council may arrange.

Section 2. A Semi-Annual Meeting may be held at such time and place as the Council may elect.

Section 3. Special meetings of the Society may be called at any time at the discretion of the Council or shall be called by the President, at a location determined by the Council, upon written request from the Boards of Governors of a majority of the Chapters. The notices for such meetings are to be sent out by the Secretary at least thirty (30) days before the date of the meeting, and are to state the business for which such meeting is called and no other business shall be entertained or transacted at that meeting.

ARTICLE C-VIII—Committees

Section 1. Council committees, nominating committee, general and special committees shall be appointed or elected as provided in the By-Laws.

ARTICLE C-IX—Election of Officers

Section 1. The President, First Vice-President, Second Vice-President, Treasurer and four (4) members of the Council shall be elected annually by letter ballot, as further provided in the By-Laws.

Section 2. The term of all elective Officers shall begin on the adjournment of the Annual Meeting of the Society. An Officer shall continue in office until his successor has been elected and installed.

Section 3. The Council at its first meeting after the Annual Meeting shall appoint a Secretary of the Society for one (1) year. The Secretary shall be subject to removal for cause by affirmative vote of two-thirds of all of the members of the Council by written ballot.

ARTICLE C-X—Local Chapters

Section 1. Local Chapters of the Society, composed of Members of all grades may be organized at the discretion of the Council. Such Local Chapters shall operate in accordance with the Constitution, By-Laws and Rules of the Society.

ARTICLE C-XI—Funds

Section 1. The collection, deposit, disbursement and investment of all funds of the Society, except the funds of the Research Laboratory, shall be subject to the direction of the Council.

ARTICLE C-XII—Publications and Papers

Section 1. The publications and papers of the Society shall be issued subject to the direction of the Council.

ARTICLE C-XIII—Research

Section 1. All research activities of the Society shall be conducted by the Committee on Research in accordance with the Regulations governing the Committee on Research of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, which Regulations are attached hereto as APPENDIX A.

ARTICLE C-XIV—Society's Endorsement

Section 1. Recommendation, endorsement or approval by the Society or the Council shall not be given to, or made for any individual, partnership, corporation or association nor of, or for any scientific, literary, mechanical or engineering production, other than as affecting public welfare, and when affecting public welfare the opinion so expressed shall not convey any intent to promote the interests of any individual, partnership, corporation or association, nor shall such expression be harmful to any other similar interests, nor shall it differentiate between any two similar methods, processes, devices or apparatus by special commendation of one over the other.

ARTICLE C-XV—Professional Practice

Section 1. The Society will not allow its name, initials or insignia to be used in any commercial work, except to indicate conformity with its standards in accordance with the By-Laws and Rules.

Section 2. In all professional practice or business relations the members of the Society shall conduct themselves according to the CODE OF ETHICS incorporated in the By-Laws.

ARTICLE C-XVI—Amendments

Section 1. At any meeting of the Society any person entitled to vote may propose in writing an amendment to this Constitution, provided such amendment bears the written endorsement of at least ten (10) members entitled to vote. Such proposed amendment shall not be voted on for adoption at that meeting, but shall be open to discussion and modification and to a vote as to whether, in its original or modified form, it shall be mailed to the Members of the Society for action. If, by a majority vote, not less than fifteen (15) Members voting in favor thereof, it is decided to submit the amendment to the Membership for action, the Secretary shall mail in printed form to each Member entitled to vote, at least sixty (60) days prior to the next Annual Meeting of the Society, a copy of the proposed amendment as decided by said vote, accompanied by any comment the Council may choose to make.

Section 2. A ballot shall be enclosed with the proposed amendment, and the voting shall be by sealed letter ballot, closing at noon of the tenth (10th) day preceding the Annual Meeting. The ballots shall be voted and tabulated as provided in the By-Laws and Rules. The result shall be decided by a majority of the votes cast. At the Annual Meeting of the Society following the closure of the voting, the presiding officer shall announce the result of the vote, and if the amendment is adopted it shall thereupon take effect.

Section 3. Any changes in the order of consecutive numbering of existing articles or sections of the Constitution, made necessary by such adopted amendment, shall be authorized by the Council.

BY-LAWS

ARTICLE B-I—Objects and Government

Section 1. To accomplish the objects of the Society, its activities shall include:

(*a*) Holding meetings for reading and discussing professional papers, and the interchange of knowledge and opinions afforded by personal contact with our fellow men.

(*b*) Issuing publications.

(*c*) Co-ordinating and conducting fundamental and practical research work for the benefit of the public and the engineer, and to produce the data upon which codes may be based.

(*d*) Encouraging the production of data, reports, standards, and codes.

Section 2. The headquarters of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS shall be located in the City of New York, N. Y., at such address as the Council may select.

Section 3. The majority of votes cast shall decide every question coming before any meeting of the Society or the Council or a Committee, unless otherwise provided in the Constitution, the By-Laws, the Rules or the laws of the State of New York.

Section 4. The Society may copyright any of its data, reports or publications, or patent any of its research developments, at the discretion of the Council.

Section 5. The Council may authorize the co-operation of the Society with other organizations, if such action in its opinion seems desirable.

Section 6. The Council shall have full authority to conduct all business transactions for the Society except as limited herein by the Constitution, By-Laws and Rules.

Section 7. Whenever the Council or two-thirds of the Members present and voting at any Annual or Semi-Annual Meeting refer any question, except an amendment to the Constitution, to the membership at large, it shall be submitted in writing, in such form as directed by the Council, to all members entitled to vote and decided by a majority of the letter-ballots received.

Section 8. A quorum for the transaction of business at any meeting of the members shall consist of not less than forty (40) members entitled to vote.

Section 9. At any regular meeting, the Council may by a two-thirds vote of its members present, adopt or amend the Rules in harmony with the Constitution and By-Laws, provided that such Rule or Amendment shall have been submitted in writing to each member of the Council at least two (2) weeks before the meeting at which action is to be taken. A Rule or an Amendment to a Rule shall take effect immediately upon its adoption by the Council, and shall be published in the next issue of the JOURNAL.

Section 10. The Regulations which govern the Committee on Research shall be subject to addition, amendment or repeal by a majority of the members present and voting at any meeting of the Society, provided that a copy of such proposed addition, amendment or repeal shall have been submitted in writing to all members entitled to vote at least thirty (30) days prior to the time at which action is to be taken.

ARTICLE B-II—Membership

Section 1. Any Member whose dues are paid in full may resign at any time. Resignations must be presented in writing addressed to the Society at its headquarters. The

Secretary shall deliver all resignations to the Council and action must be taken at the first meeting of the Council following receipt of same.

Section 2. Any Member whose dues shall remain unpaid for one (1) year shall at the discretion of the Council forfeit the privileges of membership, and if he neglects or refuses to pay his dues within thirty (30) days after notification from the Secretary, his name shall be stricken from the roll of members.

Section 3. The Council, by a two-thirds vote, may expel any Member of any grade who may be adjudged to have violated the Constitution or By-Laws of the Society, or who shall have been found guilty, after an adequate opportunity of a hearing, of conduct rendering him unfit to continue in its membership.

Section 4. Any person ceasing to be a member of the Society, through resignation or otherwise, shall forfeit all right, title and interest in the property of the Society.

Section 5. Any person who has been a member of the Society for fifteen (15) years or more and has retired from business, then upon reaching the age of seventy (70), shall have his dues remitted for the current year and for ensuing years, without surrendering any of the privileges of membership as long as he lives.

ARTICLE B-III—Admission and Advancement

Section 1. A candidate for admission to any grade of membership in the Society, except Honorary Membership, must make application on a form approved by the Council, upon which he shall write a statement giving a complete account of his qualifications and engineering experience, and an agreement that he will, if elected, conform to the Constitution, By-Laws and Rules of the Society.

Section 2. A candidate for any grade of membership in the Society, except Honorary Membership, must be proposed by two (2) members to whom he must be personally known, except as provided in *Section 3*, and such application must be seconded by two (2) other members.

Section 3. In case applicants for membership are not acquainted with members of the Society, endorsements from the faculty of educational institutions, or the recommendations of the members of other professional societies, or by the officers of responsible corporations, who are well acquainted with the applicant, may be considered, if sufficient evidence has been submitted to satisfy the Admission and Advancement Committee that the applicant is worthy of admission to membership.

Section 4. The proposers of a candidate may be required to submit a complete account of the qualifications of the candidate, including a statement in writing of the professional experience and the character and diversity of the work performed, age, education and any other qualifications the Council may demand.

Section 5. Honorary Membership shall be granted at an Annual Meeting of the Society only and all such proposals shall have the unanimous endorsement of the Council before they are submitted to the meeting for election.

Section 6. All applications for membership are to be addressed to the Society at its headquarters and the names of applicants and their references shall be printed in the next issue of the JOURNAL or sent to the members in such other manner as may be ordered by the Council.

Section 7. When the Admission and Advancement Committee has acted favorably upon a Candidate's application and assigned his grade the Council shall vote upon the election of the proposed Candidate for membership by letter-ballot.

Section 8. If two-thirds of the Council shall cast a vote favorable to a candidate he shall be elected, and his name shall be published in the next issue of the JOURNAL.

Section 9. The names of the candidates, who are not elected, shall neither be recorded nor announced in the proceedings of the Council.

Section 10. Any person having been elected to membership in the Society shall be promptly notified by the Secretary, whereupon he shall accept such election, subscribe to the Constitution, By-Laws and Rules, and pay the amount of his dues within three (3) months after such notice of election shall have been sent him, or his election shall become void. Failure to qualify within the time allowed will mean forfeiture of the Initiation Fee.

Section 11. Any member of the Society is entitled to membership in the local Chapter nearest to his residence or place of business, provided he pays his Chapter dues in accordance with the Constitution and By-Laws of that Chapter.

ARTICLE B-IV—Admission Fees and Dues

Section 1. The admission fee of Members, Associate Members, Junior Members and Student Members shall be as determined by the Council until 1935 and thereafter the admission fee of Members and Associate Members shall be fifteen dollars ($15.00); of Junior Members five dollars ($5.00); and of Student Members two dollars ($2.00). Admission fee must accompany application.

Section 2. The annual dues of Members, Associate Members, Junior Members and Student Members shall be as determined by the Council until 1935 and thereafter the dues of Members and Associate Members shall be twenty-five dollars ($25.00); of Junior Members twelve dollars ($12.00); and of Student Members three dollars ($3.00).

Section 3. Honorary Members shall be exempt from payment of admission fee and dues.

Section 4. Of the annual dues paid by the members of each grade, three dollars ($3.00) shall be considered as a subscription for the JOURNAL of the Society.

Section 5. Of the annual dues paid by Members and Associate Members forty per cent (40%) shall be considered as a direct contribution to the Research Fund and shall be immediately deposited in said fund and shall not be used for any other purpose.

Section 6. All dues shall be payable in January of each year in advance. The dues of new members of all grades, shall be due and payable on the first day of the month following the date of admission of such members.

Section 7. The dues of a new member of any grade may be pro-rated monthly for the balance of the year but if the amount thus paid is less than five dollars ($5.00) such member shall not be entitled to receive the volume of the TRANSACTIONS for the year in which he is elected, but he shall otherwise be entitled to all the rights and privileges of membership. Forty per cent (40%) of the pro-rated dues of Members and Associate Members shall be considered as a contribution to the Research Fund and shall be immediately deposited in such fund and shall not be used for any other purpose.

Section 8. Junior Membership shall be limited to ten (10) years and upon election to a higher grade, such members shall upon notification of transfer, pay an additional fee of ten dollars ($10.00) and thereafter pay the annual dues of the grade to which they are transferred.

Section 9. Student Members upon election to a higher grade, shall upon notification of transfer pay the annual dues of the grade to which they have been transferred.

Section 10. The Council may at its discretion restore to membership (upon such terms as its judgment dictates) any person suspended for non-payment of dues.

ARTICLE B-V—The Council

Section 1. Five (5) members of the Council shall constitute a quorum for the transaction of business at any Council Meeting.

Section 2. The Council shall hold meetings, have an office and keep the books of the Society at the office selected as headquarters in the City of New York, N. Y. Meetings of the Council may at its convenience be held at such other place or places as the Council from time to time may determine.

Section 3. Regular meetings of the Council shall be held quarterly.

Section 4. Special meetings of the Council shall be held whenever called by the President or when requested by not less than three (3) Council Members.

Section 5. The Council shall present at the Annual Meeting a complete report of the work of the Society during the preceding year.

Section 6. If any Officer of the Society or Committee Member fails in the performance of his duty, from inability or otherwise, the Council may, by a two-thirds vote, declare his office or position vacant. The Council shall then immediately appoint a member to fill the vacancy pending the next annual election of officers, except for the office of President, which shall be filled by the First Vice-President, or the office of First Vice-President, which shall be filed by the Second Vice-President.

ARTICLE B-VI—Officers

Section 1. The President shall be the chief executive officer of the Society. He shall preside at all meetings of the Society and of the Council. He shall have general charge and supervision of the business and affairs of the Society. He shall sign in behalf of, and in the name of the Society, all contracts and all certificates of membership in the Society. He shall make a report of the affairs of the Society at the Annual Meeting and shall do and perform such other duties as usually appertain to his office. He shall not be eligible for immediate re-election to the same office at the expiration of the term for which he was elected.

Section 2. The First Vice-President shall possess all the powers and perform the duties of the President in his absence or disability. He shall perform such other duties as may be assigned to him by the Council.

Section 3. The Second Vice-President shall possess all the powers and perform the duties of the President in the absence or disability of the President and First Vice-President. He shall have such other powers and perform such other duties as may be assigned to him by the Council.

Section 4. The Treasurer shall have the custody of all the funds of the Society, as provided in Article B-XI. He shall at all reasonable times exhibit his books and accounts to any member of the Council, and shall perform all duties incident to the office of the Treasurer, subject to the direction of the Council. He shall give a bond in a penal sum and with a surety or sureties approved by the Council, for the faithful performance of his duties as Treasurer. If a surety company bond is furnished the premium therefor shall be paid by the Society.

Section 5. The Secretary shall keep the minutes of all meetings of the Council and the minutes of all meetings of the Society. He shall attend to the giving and serving of all notices of the Society and of the Council. He shall sign with the President in the

name of the Society all contracts, membership certificates or official documents requiring the corporate signature and shall affix the seal of the Society to such papers or documents as require the same. He shall have charge of such books and papers of the Society as the Council may direct, all of which at all reasonable times shall be open to the examination of any member of the Society. He shall in general perform all the duties incident to the office of Secretary and shall perform such other duties as may be assigned to him by the Council. He shall give a bond in a penal sum and with a surety or sureties approved by the Council, for the faithful performance of his duties as Secretary. If a surety company bond is furnished the premium therefor shall be paid by the Society. Assistant Secretaries may be appointed by the Council.

Section 6. The Secretary shall receive a salary which shall be fixed by the Council. No other Officer of the Society shall, as such, receive any salary or compensation for his services.

Section 7. Each President of the Society, upon retirement from office shall automatically be entitled to Presidential Membership, which shall be subject to the same dues as full membership. The Presidential Members shall constitute an Advisory Council, to which the Council may refer matters of policy of the Society for advice. The Advisory Council shall be accorded the privilege of submitting suggestions on matters of policy to the Council. The retiring President shall act as chairman of such Advisory Council.

ARTICLE B-VII—Meetings

Section 1. Meetings of the Society may be held at such times and places as the Council may direct, except as provided for in the Constitution.

Section 2. Announcements of all meetings of the Society shall be published in the JOURNAL and the Secretary shall also mail a notice of same to each member of all grades, not less than thirty (30) days before the date of each meeting.

Section 3. The expenses of the Annual and Semi-Annual meetings of the Society shall be paid for from funds of the Society. When a meeting is to be held in or adjacent to a city where a Chapter of the Society is located, the Council may request the assistance of the Local Chapter in arranging details of the meeting, entertainment features, etc., but the Chapter shall not solicit nor allow any solicitation of funds to help defray the expenses of such meeting or entertainment features.

ARTICLE B-VIII—Committees

COUNCIL COMMITTEES

Section 1. At the first meeting of the Council after the Annual Meeting, the President shall appoint from the members of the Council the following committees consisting of three (3) members each, to act under the direction of the Council:

(*a*) Executive Committee.
(*b*) Finance Committee.
(*c*) Membership Committee.
(*d*) Meetings Committee.

Section 2. The Executive Committee shall investigate and make recommendations to the Council regarding all matters having to do with the professional or business status of the Society and matters which might in any way reflect upon one or more members of the Society, or the Society as a whole. It shall exercise such other functions as the Council may specify.

Section 3. The Finance Committee shall have supervision of the financial affairs of the Society, except those of the Committee on Research. The Treasurer shall be an ex-officio member of this Committee.

Section 4. The Membership Committee shall cooperate with the members of the Society in an endeavor to secure the applications for membership of qualified candidates and thru local Chapters committees furnish information about the aims, activities and achievements of the Society.

Section 5. The Meetings Committee shall select from the papers which have been approved by the Publication Committee the papers for presentation at the technical sessions of the Annual and Semi-Annual Meetings, and shall assist the Publication Committee in securing desirable papers.

GENERAL COMMITTEES

Section 6. There shall be an Admission and Advancement Committee of three (3) members, appointed by the Council, for a term of three (3) years, the term of one member expiring each year. This Committee shall receive from the Secretary all applications for membership, make rigid inquiry as to the eligibility of candidates, and report to the Council only such as have been approved. In case of disapproval, only the proposers and the applicant shall be notified of such action. The proceedings of the Committee shall be private and confidential. It shall be the duty of this Committee each year to recommend to the Council, for transfer of grade of membership all Junior Members who have reached the age of thirty (30).

Section 7. There shall be a Publication Committee of three (3) members, appointed by the Council, for a term of three (3) years, the term of one member expiring each year. This Committee shall receive and examine all papers intended for presentation to the Society and accept such as it may approve. This Committee shall review the papers and discussions which have been presented at the meetings and shall decide what papers and discussions, or parts of the same, shall be published. It shall approve for publication the TRANSACTIONS of the Society, containing the papers, discussions and abstracts of the minutes of the Society and of the Council. No member shall publish any papers as having been read before the Society without obtaining the consent of this Committee, and such consent shall not be construed to be an endorsement by the Society of any statements contained in such papers or publications.

Section 8. Committee on Research. There shall be a Committee on Research of fifteen (15) members, which shall function in accordance with the Regulations which govern the Committee on Research. The Council shall nominate previous to July first of each year, five (5) members of the Society, to serve for three (3) years, to replace the five (5) members of the Committee on Research whose terms expire at the close of the Annual Meeting of the Society following such nomination. The Council shall vote on the nominees by letter ballot, and the Secretary shall publish the names of the candidates nominated in the October issue of the JOURNAL.

Section 9. Constitution and By-Laws Committee. At the first meeting of the Council after the Annual Meeting, the President shall appoint a Committee on Constitution and By-Laws, which Committee, under the direction of the Council, shall have supervision of all matters affecting the Constitution, By-Laws and Rules. This Committee shall consist of three (3) members, and shall report to the Council in writing on all matters referred to it by the Council.

Section 10. The Nominating Committee shall consist of one (1) member eligible to vote designated by each Chapter, or his alternate also appointed by the Chapter. The

Secretary of each Chapter shall certify to the Secretary of the Society on or before January first the names of the member and alternate selected.

The Committee shall meet at the Annual Meeting of the Society at the call of the Secretary of the Society and shall effect its own organization and elect its own Chairman. At the Semi-Annual Meeting of the Society, if possible, the Nominating Committee shall select the nominees for the ensuing year for the offices of President, First Vice-President, Second Vice-President, Treasurer, and four (4) members of the Council. In any event the names of the nominees shall be certified to the Secretary of the Society before September twentieth, with the written consent of each nominee to fill the office for which he has been selected and their names with the offices to which they have been nominated shall be published in the October issue of the JOURNAL.

Section 11. At the organization meeting of the Council, each year, the president shall appoint a Committee of five (5) members to be known as the F. Paul Anderson Award Committee. This Committee shall function according to the terms of the F. Paul Anderson Award. (See APPENDIX B.)

Section 12. The Council shall have power to delegate to any other committee any duties that may be deemed necessary or expedient in the interests of the Society and such committee shall report to the Council when and as required. The Council shall also appoint any committee and direct its proceedings as may be ordered by resolution of the Society at any meeting.

ARTICLE B-IX—Election of Officers

Section 1. Twenty (20) or more members of the Society, eligible to vote, may present to the Secretary, over their signatures, the name of any member eligible to hold office in the Society as a candidate for any office, provided such name is presented at least sixty (60) days preceding the next Annual Meeting, together with the written consent of the nominee to fill the office for which he has been selected, and the Secretary shall add such names to the ballot if they are not already included in the list of names presented in the formal report of the Nominating Committee. Such names when presented shall be included on the printed ballot, with special notation that they are presented by members independent of the Nominating Committee report. Any member nominated for any office may withdraw his name as a candidate, whether nominated by the regular Nominating Committee or by twenty (20) or more members, providing he does so at least forty-five (45) days prior to the next Annual Meeting.

Section 2. The Secretary shall prepare ballots with the names of all candidates and forward them to the members, eligible to vote, at least thirty (30) days before the date of the Annual Meeting.

Section 3. Each member entitled to vote shall cancel the names of all candidates for whom he does not wish to vote and return his ballot so that it will reach Society headquarters before noon of the tenth (10th) day preceding the Annual Meeting. A member may write upon his ballot the name of any member for whom he wishes to vote, if such name is not on the printed ballot. The ballot is to be enclosed in a blank envelope which shall in turn be enclosed in another one endorsed by the voter.

Section 4. The Secretary shall prepare a list of the members who have not paid their dues within the year preceding the Annual Meeting, and no vote shall be counted for any member whose name is endorsed on the outer envelope if such name shall appear in the list of delinquents.

Section 5. The ballots shall be opened and the result of the vote declared on the first day of the Annual Meeting by three (3) tellers appointed by the President. The

candidates receiving the greatest number of votes for the several offices shall be declared elected, and shall take office at the close of the last session of the Annual Meeting.

Section 6. In the event of a tie vote at any election of officers of the Society, the Council, by a majority vote, shall decide the tie.

ARTICLE B-X—Local Chapters

Section 1. (*a*) Local Chapters of the Society may be formed upon application of ten (10) members if the organization of such local Chapter will, in the judgment of the Council of the Society, advance the Society's interests.

(*b*) Upon recommendation of the Council, a charter may be granted to form such local Chapter, which shall be operated and conducted under the control and at the pleasure of the Society; such local Chapter shall be governed by the Constitution, By-Laws and Rules of the Society and by such other local By-Laws as may be adopted by the local Chapter and approved by the Council of the Society before becoming operative.

(*c*) The membership of such Chapter shall comprise only members of the different grades in good standing in the Society; any member of any local Chapter who shall cease to be a member of the Society, shall thereby forfeit all right to membership in such local Chapter.

(*d*) Every such local Chapter when formed shall be chartered in the name of the State, section of State, County or City, in which the same shall be located.

(*e*) Student Branches of the Society may be established, under the direction of the Council.

Section 2. Any person delivering an address before any Chapter of the Society shall be requested not to issue it for publication without first securing the approval of the Publication Committee of the Society.

Section 3. The charter of a local Chapter may be revoked, for cause, by majority vote of Council.

ARTICLE B-XI—Funds

Section 1. When so directed by the Council, the Chairman of the Finance Committee shall invest, such portion of any funds of the Society as determined by the Council, in securities of the United States Government. All investments shall be approved by the Council.

Section 2. Any bequest or gift of moneys to the Society which the donor shall designate to be used for a specific purpose shall, after acceptance by the Council, be deposited or invested in the manner provided by *Section 1*, and the income or principal, as designated by the donor, used for the specific purpose designated.

Section 3. An endowment fund for research and such other purposes devoted to the art of heating and ventilating and air conditioning as may be determined by the Council shall be established. The interest or income from this fund shall be used each year as shall be determined by the Council. The principal shall remain intact and shall be deposited in banks or invested in securities of the United States Government.

Section 4. At the organization Meeting of the Council of each year the Finance Committee shall present to the Council a budget of estimated income and expenditures for the current year, which after approval by the Council shall govern the expenditure of Society funds for that year. Any proposed expenditure of Society funds outside of the approved budget shall be approved by the Council before the expenditure is made.

Section 5. Any money due the Society shall be collected by the Secretary, who shall enter all receipts in the books of the Society and deposit same to the Treasurer's account. The Secretary shall have the authority to pay salaries, traveling expenses and petty cash in accordance with the budget. In case of disability or absence of the Treasurer, the Chairman of the Finance Committee is authorized to sign checks. He shall give bond in the same manner as provided for the Treasurer.

Section 6. After December thirty-first, and before the Annual Meeting of the Society in January, the accounts of the Society shall be audited by a certified public accountant, selected by the Council at its last meeting in the calendar year. The auditor's report shall be presented at the Annual Meeting of the Society by the Chairman of the Finance Committee, and published in the Society JOURNAL.

Section 7. The current Research funds shall be handled separately from the funds of the Society, in accordance with the By-Laws of the Society and the Regulations which govern the Committee on Research. Surplus Research funds shall be invested by the Council in accordance with *Section 1*.

Section 8. A Society Reserve Fund shall be created into which all admission fees and such other moneys as the Council may direct shall be placed until the Fund totals a sum equal to fifteen dollars ($15.00) per member. This Reserve Fund is to be used only in cases of emergency when current Society revenues are insufficient to pay necessary expenses. Withdrawals not to exceed twenty per cent (20%) of the fund in any calendar year may be authorized by the Society at any meeting provided it has been recommended by the Council. Investments shall be made in accordance with *Section 1* of **Article** B-XI. Interest earned on the Reserve Fund shall be added to current Society income.

ARTICLE B-XII—Publications and Papers

Section 1. A JOURNAL carrying papers approved by the Publication Committee, news items of Society interest and such other items as may be decided by the Council shall be published monthly.

Section 2. The Society shall publish a GUIDE, as directed by the Council, containing technical information, an advertising section, and such other items as may be decided upon.

Section 3. The TRANSACTIONS of the Society for each year shall be published as soon as is practicable after the Semi-Annual Meeting.

ARTICLE B-XIII—Research

Section 1. The work conducted by the Research Laboratory of this Society shall be confined to a determination of the basic or fundamental principles or laws underlying all matters in the science of heating, ventilating or air conditioning.

ARTICLE B-XIV—Society's Endorsement

Section 1. The Society may receive papers and reports on experiments, improvements and developments of every character affecting the arts and sciences of heating and ventilating and air conditioning and it may freely discuss same and have the proceedings published in the TRANSACTIONS of the Society, without any direct or implied endorsement by the Society of such papers or reports.

Section 2. As Society research work progresses and reports are made to the Society such reports are not to be construed as an endorsement or repudiation by the Research Laboratory or by the Society, unless a specific statement of endorsement or repudiation is included.

ARTICLE B-XV—Professional Practice

Section 1. All members of the Society shall subscribe to the following Code of Ethics as required by the Constitution:

CODE OF ETHICS FOR ENGINEERS

Engineering work has become an increasingly important factor in the progress of civilization and in the welfare of the community. The engineering profession is held responsible for the planning, construction and operation of such work and is entitled to the position and authority which will enable it to discharge this responsibility and to render effective service to humanity.

That the dignity of their chosen profession may be maintained, it is the duty of all engineers to conduct themselves according to the principles of the following Code of Ethics:

1. The engineer will carry on his professional work in a spirit of fairness to employees and contractors, fidelity to clients and employers, loyalty to his country and devotion to high ideals of courtesy and personal honor.
2. He will refrain from associating himself with or allowing the use of his name by an enterprise of questionable character.
3. He will advertise only in a dignified manner, being careful to avoid misleading statements.
4. He will regard as confidential any information obtained by him as to the business affairs and technical methods or processes of a client or employer.
5. He will inform a client or employer of any business connections, interests or affiliations which might influence his judgment or impair the disinterested quality of his services.
6. He will refrain from using any improper or questionable methods of soliciting professional work and will decline to pay or to accept commissions for securing such work.
7. He will accept compensation, financial or otherwise, for a particular service, from one source only, except with the full knowledge and consent of all interested parties.
8. He will not use unfair means to win professional advancement or to injure the chances of another engineer to secure and hold employment.
9. He will co-operate in upbuilding the engineering profession by exchanging general information and experience with his fellow engineers and students of engineering and also by contributing to work of engineering societies, schools of applied science and the technical press.
10. He will interest himself in the public welfare in behalf of which he will be ready to apply his special knowledge, skill and training for the use and benefit of mankind.

ARTICLE B-XVI—Amendments

Section 1. The By-Laws of the Society shall be subject to addition, amendment or repeal by a majority of the members present and voting at any Annual or Semi-Annual Meeting of the Society, or at any special meeting of the members called for that purpose, provided that a copy of such proposed addition, amendment or repeal, endorsed by at least ten (10) members of the Society eligible to vote, shall have been submitted in writing by the Secretary, to all of the members entitled to vote, at least thirty (30) days prior to the time of the meeting at which such addition, amendment or repeal is to be considered.

RULES

ARTICLE R-I—Objects and Government

Section 1. All questions arising at any meeting involving parliamentary rules not provided for in these By-Laws, shall be governed by *Roberts' Rules of Order, Revised.*

ARTICLE R-II—Membership

Section 1. Each member upon election shall be furnished, without charge, a Certificate of Membership signed by the President and Secretary of the Society.

Section 2. Certificates of Membership shall be furnished framed, if desired, upon payment of four dollars ($4.00).

Section 3. The emblem pin of the Society shall be furnished members desiring it upon payment of two dollars ($2.00).

Section 4. Upon termination of membership a member shall return to the Society the certificate of membership which was issued to him, and also the emblem pin if he has one. A refund of one dollar ($1.00) will be made by the Secretary upon the return of the emblem pin.

ARTICLE R-III—Admission and Advancement

Section 1. Immediately upon receipt of an application for membership, the Secretary shall send it with the information forms submitted by each proposer and seconder, to the Admission and Advancement Committee which will determine the applicant's grade of membership.

Section 2. The Admission and Advancement Committee may obtain information regarding any applicant from members other than the proposers and seconders.

Section 3. When the Admission and Advancement Committee has reported its recommendations as to the grade of membership awarded each applicant, the secretary shall mail to each member of the Council a ballot carrying the names and grades of membership recommended. It shall be the duty of each Council member to vote and promptly return the ballot to Society headquarters.

ARTICLE R-IV—Admission Fee and Dues

Section 1. The Secretary shall promptly return the admission fee to a candidate for membership whose application is unfavorably acted upon.

Section 2. That portion of the dues collected from Members and Associate Members allocated to research, shall be transferred quarterly to the Committee on Research.

ARTICLE R-V—The Council

Section 1. The Order of Business at any Regular Meeting of the Council shall be as follows:

(*a*) Report of Committees.
(*b*) Miscellaneous Business.
(*c*) Motions and Resolutions.

ARTICLE R-VI—Officers

Section 1. During his term of office the President shall visit as many Chapters of the Society as his convenience permits.

Section 2. The Officers of the Society shall endeavor to assist the Chapters to promote the interests of the Society.

ARTICLE R-VII—Meetings

Section 1. The order of business at the Annual Meeting of the Society shall be as follows:

(*a*) Report of Officers.
(*b*) Reports of Committees.
(*c*) Report of Tellers of Annual Election.
(*d*) Unfinished Business.
(*e*) Reading and Discussion of Papers.
(*f*) New Business.
(*g*) Installation of Officers.
(*h*) Adjourn.

Section 2. Before July first of each year the Council shall select the location of the next Annual and Semi-Annual Meetings of the Society. The location of succeeding meetings may be determined at any meeting of the Society by a majority vote of those present.

Section 3 (*a*) At professional sessions each paper shall be read in abstract, ten (10) minutes being allowed to the Author for the presentation, unless otherwise authorized by the meeting.

(*b*) A member who has given notice of his intention to discuss a paper and has reduced his discussion to writing, may be allowed ten (10) minutes for its presentation.

(*c*) Each speaker shall be limited to five (5) minutes in the oral discussion of a paper, unless the time shall be extended by action of the meeting. A member who has once had the floor, may not again claim it, until all others who desire to speak on that paper have been heard.

(*d*) Authors of papers may have five (5) minutes to close the discussion on the paper.

Section 4. Six (6) months in advance of the date of an Annual or Semi-Annual meeting the Secretary shall call the attention of the President of the Local Chapter of the place where the meeting is to be held to Article B-VII, *Section 3*, of the By-Laws, and if this section of the By-Laws is violated the Secretary shall report same to the Council.

Section 5. At any meeting of the Society no exhibit of equipment and no advertising displays will be allowed in rooms or halls occupied by the Society or in rooms or halls adjacent thereto.

ARTICLE R-VIII—Committees

The following does not apply to the Committee on Research which is governed by its own Regulations.

Section 1. The President of the Society shall appoint the Chairman and the members of all committees, and the Chairman after communicating with the other members of the committee shall fix the time and place of the first meeting. He may at his discretion request one or more members to prepare material in advance for consideration of the committee.

Section 2. When any member of a committee shall have correspondence with another member of that committee, copies of such correspondence shall be sent to all members of the committee and to the Secretary of the Society. All copies shall indicate that copies have been sent to all committee members and the Secretary of the Society.

Section 3. The meetings of any committees shall be open only to its own membership and to such visitors as may be approved by the Chairman.

Section 4. Each standing or special committee thru its Chairman shall render a report at the Annual or Semi-Annual Meeting of the Society as instructed by the Council. The term of office of members of all appointed committees shall end at the close of each Annual Meeting, and any member shall be eligible for re-appointment.

Section 5. Expense for postage incurred with the business of authorized committees will be refunded by the Secretary of the Society on presentation of vouchers approved by the Chairmen of committees. All correspondence relating to the business of committees may be conducted on official stationery of the Society, which shall be furnished by the Secretary upon request. Items of expense other than postage will not be assumed by the Society unless such expenditures were incurred in pursuance of previous authorization by the Council of the Society and within the limits specifically fixed by the Council.

Section 6. Committees are not authorized to pay salaries or professional fees in any form to any of their officers or members.

Section 7. Any appointed committee may be discharged by the Council either at its own request or with its consent upon completion of the work for which it was appointed, or in consequence of protracted inactivity.

Section 8. Any appointed committee may be discharged for cause by the Council upon its own initiative.

Section 9. Code for Publication Committee.

Object of Code: To facilitate work and render policy more permanent.

General Policy:

1. The advancement of the theory and practice of heating, ventilating, air conditioning and of the allied arts and sciences.
2. The development of the individual engineer.
3. The maintenance of a high standard of achievement for the Society and its members.

Sources for Papers:

1. Members.
2. Non-Members.
3. Research Laboratory.
4. Chapters.
5. Committees.

Solicitation of Papers:

The solicitation of papers should begin immediately after, or at the time of the Annual Meeting in January.

The New Chairman:

The new Chairman should canvass all the unpublished papers from the previous year and get the advice of the outgoing chairman on all such cases. Particular attention should be given to the elimination of all manuscripts with a purely or obvious commercial bias, in fact prospective authors should be warned (if there is any risk of such a thing) that such bias will not be tolerated.

Routing of Papers (after receipt from Secretary):

Each author shall be required to submit four copies of his paper.

1. Received by Members of Publication Committee.

2. Chairman receives comments of other two members.
3. Returned to Secretary if poorly prepared or unsuitable.
4. Submitted by Chairman of Publication Committee to at least two (2) members who are specialists in subject if paper is highly specialized or theoretical. Advise with Chairman of Technical Committee as to members qualified for such review.
5. Returned by Chairman of Publication Committee to Secretary with recommendations.
6. Returned to author by Secretary if unsuitable or if in need of revision.
7. Preparation, by Secretary, for publication if acceptable.

Secretarial Action: On advice of Chairman of Publication Committee, the Secretary shall list each paper returned to him as follows:

1. For presentation at Annual or Semi-Annual Meeting.
2. For presentation at a Chapter Meeting.
3. For publication in the JOURNAL.
4. For return to author for revision or as unsuitable (definite reasons to be given for rejection).

Characteristics Desired for Papers:

1. Breadth of vision.
2. Quality.
3. Originality.
4. Scope within branches of the art or science.
5. Diversity as to subjects for meetings or publication.
6. Brief and informative summaries.
7. Clear, readable charts, graphs, drawings and photographs with all trade names eliminated.

Branches of Art or Science:

1. Heating.
2. Ventilation.
3. Humidification.
4. Dehumidification.
5. Drying.
6. Cooling as applied to drying.
7. Refrigeration as applied to drying.
8. Evaporation as applied to drying.

Papers Acceptable (when within scope):

1. Mathematical processes.
2. Researches.
3. Developments.
4. Design.
5. Engineering or economic investigation.
6. Functioning of equipment or systems.
7. Testing of equipment or systems.
8. Applications to industrial or social purposes.
9. Education.
10. Standardization.
11. Co-operative activities.
12. Ethical or psychological aspects.

Preference Given Papers:

1. New theoretical or practical developments.
2. Operation, design or practice.
3. Current engineering of practical nature (should be handled in symposium or reports).

Technical Committee Relations:

1. Chairman of Technical Committee is unofficial adviser to chairman of Publication Committee.

Annual and Semi-Annual Meetings Duties:

1. Assist Meeting's Committee with technical program for each Meeting.
2. Aid Secretary in preparing each Meeting papers.
3. Prepare accessory equipment for presentation of papers.
4. Advise Secretary and Presiding Officer on preference to be given in written discussions of papers.
5. Confer with local meeting committee as to subjects for papers.
6. Provide Secretary each month with a prospective list of subjects or papers which are likely to become available for various purposes.

ARTICLE R-IX—Election of Officers

Section 1. Each ballot shall have the following information printed on it:

(*a*) Date the ballot closes.

(*b*) Leave ballot unmarked if you wish to vote for all candidates as nominated. (Note:—If candidates other than those selected by the regular Nominating Committee are nominated, this paragraph is to read:—"Mark a cross (x) opposite the names of those candidates for whom you wish to vote.")

(*c*) Cross out the name of any candidate you wish to reject, and write in the name of any eligible member of the Society for whom you wish to vote.

(*d*) Enclose ballot in blank envelope and seal it.

(*e*) Enclose the blank envelope in the larger envelope, seal it and write your name thereon, in the space provided, and mail to Society's headquarters.

Section 2. No ballot received after the stated time of closure is to be counted.

Section 3. The Secretary shall certify to the competency and signature of all voters.

Section 4. The Tellers shall open and destroy the outer envelopes, and then open the inner envelopes and tabulate the results.

Section 5. No ballot without the autographic signature of the voter on the outer envelope shall be considered by the Tellers.

Section 6. In counting the ballots, the Tellers shall consider a vote for any Officer as valid providing the intent of the voter as to that particular office is clear, even though the ballot as to candidates for other offices may be invalid.

ARTICLE R-X—Local Chapters

Section 1. When a group of Society members desire to organize a Chapter of the Society, they shall state in their application to the Council the location where it is desired to establish headquarters and the boundaries which in their opinion it would be desirable to include.

Section 2. Upon approval of a petition the Council will, if requested, delegate a member of the Society to assist in the formation of the new Chapter.

Section 3. A member of the Society may not hold office in more than one Chapter at the same time.

ARTICLE R-XI—Funds

Section 1. The Secretary shall receive all bills against the Society, and shall present them for audit and approval to the Chairman of the Finance Committee. The approved bills shall be referred to the Treasurer, which officer, if he also approves the bills, shall draw and sign a check payable to the account of AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS for the total amount of the approved bills. The Treasurer shall present the approved bills with the check to the President of the Society for his examination. After approval of the bills by the President, that officer shall countersign the check in payment thereof, which check shall be deposited to the account of AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and known as the Secretary's account. The Secretary shall promptly draw against this account in settlement of the approved bills.

Section 2. Each year the remaining net profits from THE GUIDE, after deducting the expense funds allotted for Annual and Semi-Annual Meetings, and proper proportionate overhead, including compensation to employees, shall be donated to the Research Fund.

Section 3. Moneys allotted the Research Fund out of profits from THE GUIDE shall be paid at such time as the Council shall determine, but in no case later than twelve (12) months from the time such profits are collected.

ARTICLE R-XII—Publications and Papers

Section 1. Publications of the Society may be sold to non-members at the following prices:

JOURNAL per year (U. S. A.)	$3.00
GUIDE	5.00
TRANSACTIONS	5.00

One copy of the above to be given to members without charge. Additional copies may be sold to members at prices determined by the Council. Other publications and the above in quantity lots shall be sold to members and non-members at prices determined by the Council.

Section 2. Papers may be received by the Society by voluntary contribution or by request. All papers shall be referred to the Publication Committee for examination, and any papers may be returned to the Authors with the recommendations of the Committee for modification or changes. The papers may then be re-submitted.

Section 3. The Meeting's Committee shall select the papers for presentation at the Annual and Semi-Annual meetings of the Society, with the approval of the Council.

Section 4. Papers selected for presentation at a meeting of the Society shall, if possible, be published in the JOURNAL before the meeting.

Section 5. Papers accepted become the property of the Society and may be copyrighted by the Society. Any person wishing to reprint papers, codes, discussions or other Society publications, either in full or in part, must obtain permission for such

reproduction by applying to the Society in writing, and if permission is granted, proper credit must be given to the authors and the Society.

Section 6. Reprints of papers, codes and TRANSACTIONS may be obtained by application to the Secretary of the Society within a reasonable time after their publication, and upon payment of charges established therefor by the Council.

ARTICLE R-XIII—Research

Section 1. The nominees to the Committee on Research shall be balloted for in the same manner and at the same time Society Officers are balloted for.

ARTICLE R-XIV—Society's Endorsement

Section 1. The use of the Society's name or the reproduction of its emblem for advertising purposes in any pamphlet, magazine or book except official publications of the Society is prohibited.

Section 2. Whenever any endorsement by the Society is desired, the request shall be sent to Society's headquarters, and submitted to the Council at its next meeting, and the Council shall advise the Society if the proposed endorsement is in accordance with the Constitution.

ARTICLE R-XV—Professional Practice

Section 1. The Council shall investigate complaints submitted regarding the professional conduct of any member, and shall have power to act either to censure by letter the member who has violated the Code of Ethics if the breach is of minor nature, or may expel such member from the Society as provided in Article B-II *Section 3* of the By-Laws.

ARTICLE R-XVI—Amendments

Section 1. When voting on an amendment to the Constitution the members shall vote according to instructions which shall be printed on each ballot in accordance with *Section 2.* A blank envelope and a larger envelope shall be sent out with each ballot.

Section 2. Each ballot for an amendment to the Constitution or By-Laws or Regulations which govern the Committee on Research shall have the following information printed on it:

(*a*) Date the ballot closes.

(*b*) Leave the ballot unmarked if you wish to vote for the amendment as printed.

(*c*) Cross out the amendment if you wish to vote against it.

(*d*) Enclose your ballot in the blank envelope and seal it.

(*e*) Enclose the blank envelope in the larger envelope, seal it and write your name thereon, in the space provided, and mail to Society's headquarters.

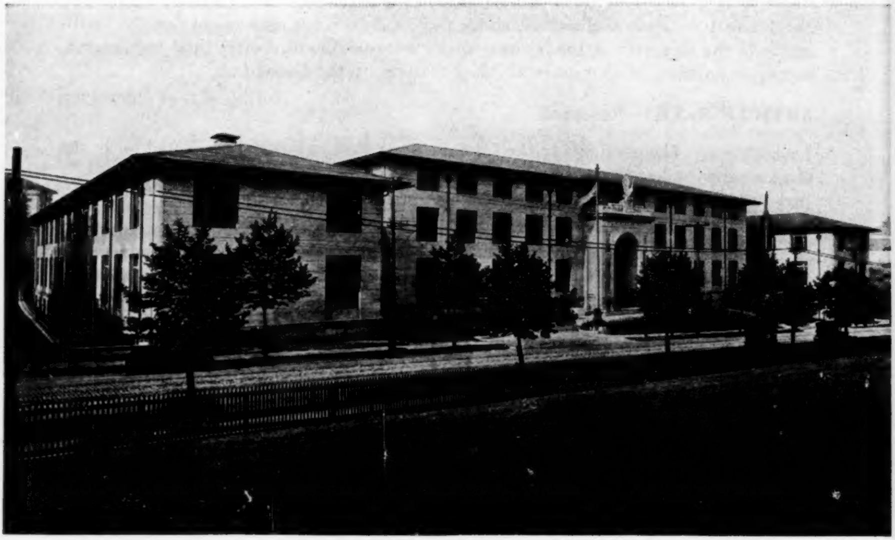
Section 3. No ballot received after the stated time of closure is to be counted.

Section 4. The Secretary shall certify to the competency and signature of all voters.

Section 5. The Tellers shall open and destroy the outer envelopes, and then open the inner envelopes and tabulate the results.

Section 6. No ballot without the autographic signature of the voter on the outer envelope shall be considered by the Tellers.

APPENDIX A



PITTSBURGH EXPERIMENT STATION OF THE U. S. BUREAU OF MINES WHERE THE RESEARCH LABORATORY OF THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS IS LOCATED

REGULATIONS GOVERNING THE COMMITTEE ON RESEARCH*

of the

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

ARTICLE I—Function

Section 1. The function of the Committee on Research is the determination and dissemination of engineering knowledge pertaining to the art of heating, ventilating and air conditioning, and the equipment and apparatus utilized by the profession.

Section 2. The purpose of the Committee on Research is to supervise the investigation, collection, tabulation and co-ordination of existing data and records of subjects pertaining to the art of heating, ventilating and air conditioning, and when the need is sufficient to warrant research or testing, to devise a plan for such procedure; also the establishment and maintenance of a Research Laboratory, and the negotiation with universities, colleges, and other organizations provided with laboratories for co-operative research and testing work.

*Originally adopted by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, January, 1919.

ARTICLE II—Organization

Section 1. Committee on Research—There shall be a standing committee known as the Committee on Research, consisting of fifteen (15) members each serving for three (3) years, and five (5) retiring each year. The outgoing Chairman if serving in that capacity during the last year of his three (3) year term on the Committee shall; without election become an additional member of the Committee on Research for one (1) year.

(*a*) The Council shall nominate previous to July first of each year five (5) members to fill the vacancies of those retiring at the next Annual Meeting.

(*b*) The nominations made by the Council shall be published in the October issue of the Society's Journal.

(*c*) Any ten (10) members of the Society eligible to vote may present to the Secretary over their signatures, the name of one (1) or more additional nominees for the Committee on Research, provided such name or names are presented at least sixty (60) days prior to the next Annual Meeting, and such additional nominations shall be placed on the ballot opposite the nominations made by the Council.

(*d*) The election shall otherwise conform to the regulations provided for the election of officers of the Society in the Constitution, By-Laws and Rules.

(*e*) Vacancies may be filled by the Council, such persons chosen by the Council to serve until a successor is elected at the next Annual Meeting.

Section 2. Chairman Committee on Research—The Committee on Research shall at its organization meeting held at the time and place of the Annual Meeting of the Society each year elect by ballot vote one (1) of their number to serve as Chairman for the ensuing year.

Section 3. Vice-Chairman—A Vice-Chairman shall also be elected in the same manner and for the same period as stated for the Chairman in *Section 2*.

Section 4. Research Executive Committee—There shall be a Research Executive Committee consisting of the Chairman of the Committee on Research and two (2) other members of the Committee on Research appointed by the Chairman immediately following the Annual Meeting.

Section 5. Research Finance Committee—At the organization meeting of the Committee on Research, or as soon thereafter as possible, the Chairman of the Committee shall appoint a committee of five (5) members of the Society to serve on the Research Finance Committee for the ensuing year. The Chairman of the Committee on Research shall appoint the Chairman of the Research Finance Committee.

Section 6. Director—There shall be a Director of the Research Laboratory.

Section 7. Technical Adviser—At the discretion of the Committee on Research a Technical Adviser to the Committee may be appointed. Such an appointment, if made, shall be made by the Committee on Research, and the term of appointment shall be for one (1) year.

Section 8. Technical Advisory Committees—The Chairman of the Committee on Research shall appoint such Technical Advisory Committees and designate a Chairman of each, as may be deemed advisable, to act in an advisory capacity to the Committee on Research and the Director of the Research Laboratory for specific projects under con-

sideration. At least one (1) member of each Technical Advisory Committee shall be a member of the Committee on Research. The Director of the Laboratory and the Chairman of the Committee on Research shall be ex-officio members of all Technical Advisory Committees.

ARTICLE III—Duties of Officers

Section 1. Chairman—The Chairman of the Committee on Research shall preside at all meetings of the Committee and the Research Executive Committee. He shall approve all bills against Research activities before payment is made.

Section 2. Vice-Chairman—The Vice-Chairman shall possess all the powers and perform the duties of the Chairman in his absence.

Section 3. Director of Research Laboratory—The Director of the Research Laboratory shall be in direct charge of the Laboratory under the direction of the Chairman of the Committee on Research. He shall be responsible to the Committee on Research, and shall co-operate with the Chairman of the Committee in every way possible. He shall approve all bills against Research activities before payment is made. He shall sign checks in payment of approved bills against all Research activities.

Section 4. Technical Adviser—The Technical Adviser if appointed shall be the consultant and adviser to the Committee on Research on such matters as may be submitted to him by the Chairman of the Committee. He shall be responsible to the Committee on Research, and shall co-operate with the Chairman of the Committee in every way possible.

ARTICLE IV—Duties of Committees

Section 1. Committee on Research—The Committee on Research shall have general charge of all Research activities including the making of the necessary contracts for rental or purchase of equipment or materials. This Committee shall select and engage a Director of the Research Laboratory, and such assistants of the Research Laboratory and the Committee on Research as may be required, or the authority for the selection and employment of the Laboratory assistants may be delegated to the Director of the Laboratory by authority of the Committee if the Committee so desires. The Committee shall determine all salaries, approve all co-operative agreements, and determine the order in which the subjects shall be investigated by the Research Laboratory. Any proposed expenditure of Research funds outside of the approved budget shall be approved by the Committee on Research before the expenditure is made.

Section 2. Research Executive Committee—(*a*) The Research Executive Committee shall function for the Committee on Research between meetings of the Committee on Research.

(*b*) The Research Executive Committee shall prepare budget of estimated income and expenditures of the Committee on Research for the current year, which after approval by the Committee on Research shall govern expenditures for that year.

Section 3. Research Finance Committee—The Research Finance Committee shall solicit the contribution of funds for the maintenance of Research activities. The Chairman of the Research Finance Committee shall keep the Chairman of the Committee on Research fully informed at frequent intervals as to the progress made in the work of his Committee.

Section 4. Technical Advisory Committees—Technical Advisory Committees shall act in an advisory capacity to the Committee on Research and the Director of the Laboratory on all subjects referred to them.

(*a*) The Chairman of a Technical Advisory Committee shall be responsible for obtaining the opinion and advice from each member of his Committee concerning the assigned subject. The advice on the assigned subject as to the methods of procedure, etc., shall be transmitted to the Chairman of the Committee on Research.

(*b*) In the case of a new Committee this advice shall be furnished as promptly as conditions shall warrant in order that there shall be no unnecessary delay in starting the investigations.

(*c*) Each Committee shall submit its recommendations in writing to the Committee on Research and render annually, not later than December first of each year, a written report of its activities.

ARTICLE V—Government

Section 1. Meetings of Committee on Research—A meeting of the Committee on Research shall be held at the time and place of each Annual and Semi-Annual Meeting of the Society. Special meetings may be called at the discretion of the Chairman, or by a majority vote of the Research Executive Committee. The place and date of the special meeting to be designated by the Chairman.

Section 2. Quorum—Five (5) members of the Committee on Research shall constitute a Quorum.

Section 3. Approval—(*a*) All acts of the Committee on Research other than those specifically authorized shall be subject to review by the Council annually.

(*b*) All acts, findings or reports of sub-committees shall be subject to the approval of the Committee on Research.

Section 4. Reports—The Reports of all investigations conducted at the Society's Research Laboratory or resulting from co-operative agreements shall be made available to all members of the Society through publication in the JOURNAL when authorized for publication by the Committee on Research and approved by the Publication Committee of the Society.

Section 5. Co-operative Agreements—At the discretion of the Committee on Research, co-operative agreements may be entered into with universities or colleges for the investigation of any specific subjects on the research program. When such a co-operative agreement is to be consumated, the Research Executive Committee shall prepare contracts in triplicate, stating the agreed upon terms, which contracts shall be signed by the authorized representative of the college or university and by the President and Secretary of the Society, as stated in Article B-VI, Sections 1 and 5 of the By-Laws of the Society, each party to the contract retaining one copy.

Section 6. Headings of Papers—All papers, findings or reports resulting from the work of the Committee on Research shall when published by the Society be headed as follows:

(*a*) If the paper, finding or report is the result of work at the Laboratory, in Pittsburgh, the following statement shall accompany a picture of the building: "Pittsburgh Experiment Station of the U. S. Bureau of Mines where the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS is located."

(*b*) If the paper, finding or report is the result of co-operative work with some other institution or laboratory, the following statement shall be used:

"This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in co-operation with (name of institution)."

Section 7. Payment of Bills—(*a*) All bills against Research activities shall be presented for approval by the Director of the Laboratory to the Chairman of the Committee on Research, who will present approved bills to the Treasurer of the Society. The Treasurer shall draw a check payable to the Research Laboratory of the A.S.H.V.E. for the total amount of the approved bills, which check shall be deposited in the special account where current funds of the Research Laboratory are carried.

(*b*) The Director of the Research Laboratory shall draw against Research Laboratory funds in payment of the approved bills. In the absence or disability of the Director of the Research Laboratory the Chairman of the Committee on Research is authorized to sign checks.

(*c*) The depository for current Research funds shall be selected by the Research Executive Committee, subject to the approval of the Council.

Section 8. Accounts—The books of account shall be kept at the Research Laboratory, and shall be audited each year at the time the Society books are audited and by the same certified public accountant, who examines the Society accounts. The accountants' report shall be presented at the Annual Meeting of the Society by the Chairman of the Committee on Research as the financial report on research activities for the year.

ARTICLE VI—Patents

If any discoveries are made under Co-operative Research agreements which are deemed worthy of patent application by the Committee on Research, any such applications and patents shall be owned jointly by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the University, with a share, to be determined by the Committee on Research and the University, to be owned by the person who makes the discovery. The cost of securing the patents shall be borne by these parties as their several interests appear.

APPENDIX B

REGULATIONS GOVERNING THE AWARD OF THE F. PAUL ANDERSON MEDAL*

ARTICLE I—The F. Paul Anderson Medal

Section 1. The F. Paul Anderson Medal shall be awarded to a member of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in recognition of notable scientific achievement or outstanding services performed in the field of heating, ventilating or air conditioning. There shall be no restriction on account of nationality or sex.

Section 2. The medal shall be of gold and shall be accompanied by a suitable certificate which shall recite the purpose of the medal and the specific achievement or services for which the award is made. The certificate shall be signed by the President and Secretary of the Society.

Section 3. The F. Paul Anderson Medal may be awarded annually but not oftener.

*Adopted by the Council December 2, 1931.

ARTICLE II—Committee of Award

Section 1. At the organization meeting of the Council each year the President of the Society shall announce the personnel of the Committee of Award for the F. Paul Anderson Medal.

Section 2. The Committee shall consist of five (5) members including the Chairman who shall be the first Vice-President.

Section 3. Vacancies on the Committee of Award shall be filled in the same manner as the original appointments are made.

ARTICLE III—Nominations for the Award

Section 1. Within three (3) months of its appointment, the Committee of Award shall invite members of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS to submit nominations for the Award and such invitation shall be published in the next available issue of the JOURNAL of the Society.

ARTICLE IV—Award of the Medal

Section 1. All nominations shall be carefully considered by the Committee for a period of not less than three (3) months.

Section 2. The Committee may make recommendations for the Award whether or not any nominations have been submitted.

Section 3. At the final Council Meeting of each calendar year the Committee on Award will report its choice of candidates for the Award of the medal and these with any additional nominations made by the Council are to be submitted to all Council members by the Secretary for vote by letter-ballot.

Section 4. At the same time letter-ballots, bearing the names of all nominees are to be distributed to all Council Members, with a statement as to when ballots are to be counted, which date is to be approximately two (2) weeks from date of mailing by the Secretary. Unless one (1) nominee receives a majority vote, successive balloting is to be carried on similarly until a majority favorable vote is secured for one (1) nominee. Names of nominees receiving only one (1) vote shall be dropped from succeeding ballots. The result of each balloting is to be given to all Council Members.

Section 5. Council has the power to decline to make any award for any year.

Section 6. No award may be made except in conformity with announcement made for the year.

Section 7. Presentation shall be made publicly by the President of the Society, at a time and place agreed upon by the recipient and the Officers of the Society.

ARTICLE V—The F. Paul Anderson Medal

Section 1. A gold medal to be known as the F. Paul Anderson Medal may be awarded annually to a member of the Society, together with a suitable certificate, in recognition of work done or services performed in the field of Heating, Ventilating or Air Conditioning. If funds are available, a bronze replica may be given with the medal.

ARTICLE VI—Funds

Section 1. The funds donated for this purpose are to be handled as prescribed by the By-Laws of the Society. The income, only, from such funds is to be used for purchase of medals and certificates.

ARTICLE VII—

Section 1. The conditions imposed by the original donor, are attached and are hereby made a part of these regulations.

ARTICLE VIII—Amendments

Section 1. These regulations may be amended by a majority vote of Council, provided advance notice has been sent to each Council Member stating the proposed amendment. Votes mailed by absent members and received by Council by the time of the voting shall be counted.

Philadelphia, Pa.
January 13, 1930

THE COUNCIL of the
AMERICAN SOCIETY OF HEATING AND
VENTILATING ENGINEERS.

Gentlemen:

I herewith take pleasure in presenting to the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS a certified check for the sum of One Thousand Dollars, subject to the following conditions:

1. That this sum, together with such other sums as may be added to it, be made a permanent endowment fund, the income from which shall be used to provide an annual medal of award (accompanied by suitable certificate), to some member of the A.S.H.V.E. in recognition of work done or services performed in the field of Heating, Ventilating or Air Conditioning:
2. That the Council of the A.S.H.V.E. shall in its discretion invest the principal and administer this fund, decide itself, or by special committee appointed by it, to whom the award shall be made, and shall decide and announce to the Society at least one year in advance, the conditions upon which the award will be made:
3. While it is herein stated that this is to be an annual award, the Council of the A.S.H.V.E. shall have the power to refuse to award the said medal in any year if in its opinion no work or service worthy of the honor has been performed in accordance with the conditions which the said Council has previously set forth. Should such conditions prevail the income from the fund shall be added to the principal:
4. That all decision of the Council relating to this award shall be made by a majority vote of all the members of the Council:
5. Should the A.S.H.V.E. go out of existence then its last Council duly elected, shall have the power to dispose of this endowment fund in such manner and for such purpose as may best promote the art of Heating and Ventilating:
6. The Name of this award shall be "The F. Paul Anderson Medal."

(Signed) THORNTON LEWIS

No. 943

STUDY OF THE APPLICATION OF THERMOCOUPLES TO THE MEASUREMENT OF WALL SURFACE TEMPERATURES

By A. P. KRATZ [1] (*MEMBER*) AND E. L. BRODERICK [2] (*NON-MEMBER*)
URBANA, ILL.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and conducted at the University of Illinois

This paper constitutes a presentation of results obtained from a study incident to the investigation in heating and ventilation conducted by the Engineering Experiment Station of the University of Illinois, of which Dean M. S. Ketchum is the Director. The work is carried on in the Department of Mechanical Engineering under the direction of A. C. Willard, Professor of Heating and Ventilation and Head of the Department. Acknowledgment is due to J. R. Fellows, Instructor in Mechanical Engineering, for valuable assistance in the preliminary stages of the study.

THE determination of surface film conductances, or surface co-efficients of heat transmission for building walls, involves the measurement of the temperature of the bounding surface between the wall and the air. From the experimental standpoint such measurements are difficult to make, and some uncertainty usually exists as to whether the temperature indicated is the temperature of the actual surface or has been influenced by the temperature of the material behind the surface or of the air in front of it. Furthermore, the presence of the measuring instrument may become a disturbing element causing the surface in its immediate vicinity to assume a temperature different from that of the remainder of the surface outside of the bounds of its influence.

A study of methods for measuring the surface temperature of an iron steam chest was made by F. W. Adams and R. H. Kean.[3] They found that, with the exception of a specially constructed thermocouple compensated for heat losses by means of a small electric heating element, the minimum error was obtained with a thermocouple of fine wire, having the wires inserted in a small hole

[1] Research Professor in Mechanical Engineering, University of Illinois.
[2] Research Assistant in Mechanical Engineering, University of Illinois.
[3] Measurement of Surface Temperatures, by F. W. Adams and R. H. Kean. Contribution from the Department of Chemical Engineering, Massachusetts Institute of Technology, Serial No. 173, Oct. 1926.

Presented at the 39th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Cincinnati, Ohio, January, 1933, by E. L. Broderick.

drilled about two millimeters into the surface, and held in place by a soft copper plug driven in between them, as recommended by P. D. Foote, C. O. Fairchild and T. R. Harrison.[4] Between the temperatures of 100 F and 300 F the last mentioned thermocouple read approximately 2.7 deg lower than the actual surface temperature. In this case, the cast-iron and the copper plug were electrically conducting materials and the thermocouple junction was formed across the surface between the two wires. Such a method is not applicable in the case of building walls in which the materials are usually more or less electrically insulating. The specially constructed compensated couple might be used in certain cases, but it requires manual application and could not be used where the surfaces are inaccessible or not exposed. Furthermore, the presence of this instrument would disturb the conditions under which the study of heat transmission coefficients is usually made, and would change the surface temperature from the normal value.

Two methods of installing thermocouples are commonly used for the study of surface temperatures of walls built of electrically non-conducting materials, namely, (1) to place the thermocouple, made of fine wire, on the surface and cover the junction and several inches of the leads with thin adhesive vellum, and (2) to embed the junction and several inches of the leads in a scratch made in the surface, and to seal the wires into the scratch with plaster of Paris, shellac, or some other plastic material. The wire and sealing material is worked flush with the surface. While very fine wires are preferable for both of these methods, wire as large as No. 22 B. and S. gage is sometimes used for the second.

In the case of the thermocouple placed on the surface and covered with vellum, since the wires extend the full diameter into the air, some uncertainty exists as to whether the temperature of the wire, and therefore the temperature indicated by the thermocouple, is influenced by the temperature of the air. The latter is usually quite different from the temperature of the surface. Also, the presence of the vellum disturbs the temperature of the surface, and corrections based on the conductivity of the vellum are somewhat speculative. In the case of the embedded thermocouple there is a possibility that the wire will assume the temperature of the layers below the surface, and that the indicated temperature will be different than the actual surface temperature. In both cases, if sufficient length of leads is not covered or embedded, heat conduction along the leads may affect the temperature indicated.

Some study of the use of embedded thermocouples for obtaining surface temperatures was made by A. P. Colburn and O. A. Hougen.[5] This study was made with thermocouples embedded in slots in condenser tubes by means of glycerol-litharge cement and made flush with the tube surface. Thus the wires were electrically insulated from the tube, but made good thermal contact. These thermocouples were used on the steam side of the tube. While these investigators found that this method gave results agreeing exactly with the actual surface temperature, the fact that the steam, the metal tube, the litharge cement, and the thermocouple wires were all media with relatively high thermal conductivities did not seem to guarantee that similar results would

[4] Pyrometric Practice, by P. D. Foote, C. O. Fairchild and T. R. Harrison, U. S. Bureau of Standards Technologic Paper No. 170, p. 298.

[5] Studies in Heat Transmission, by A. P. Colburn and O. A. Hougen. Bulletin of the University of Wisconsin Engineering Experiment Station Series No. 70, p. 69. Also *Industrial and Engineering Chemistry*, Vol. 22, May 1930, p. 522.

be obtained in the case of walls where the conductivities of the air, the wall and the plaster cement were all very low as compared with the conductivity of the thermocouple wires. A study of the use of thermocouples under the last named conditions, therefore, appeared to be warranted, and the objective in this investigation was a comparison of the readings of the thermocouples installed by methods (1) and (2), with the actual temperature of the undisturbed surface uninfluenced by the presence of the thermocouples themselves.

The determination of the true surface temperature was based on the premise that if a temperature gradient curve for a known homogeneous material could be established to within ⅛ in. of the surface, this curve could be extrapolated to the surface and would accurately represent the temperature that the

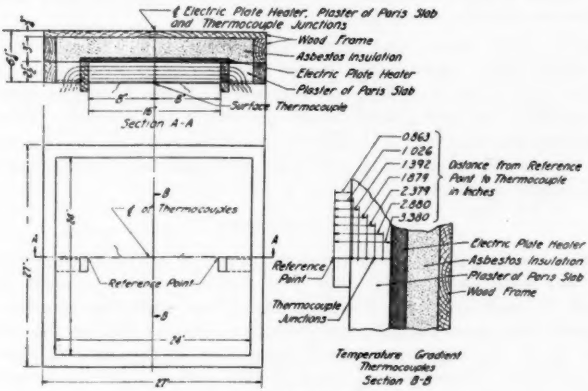


FIG. 1. TEST SLAB

undisturbed surface would assume, provided that any local disturbance did not extend into the material as far as the position of the last temperature used as a point on the temperature gradient curve.

DESCRIPTION OF APPARATUS

The principal apparatus used consisted of a homogeneous plaster of Paris slab 24 in. x 24 in. and 2½ in. thick. This slab was placed in a vertical position with one surface exposed to the air of the room as shown in Fig. 1. Five thermocouples, of No. 34 B. and S. gage copper and constantan wire, were embedded in the slab with their junctions placed in the short central axis and with the lead wires in planes parallel to the faces of the slab. The distances between these junctions measured along the short central axis and perpendicular to the face of the slab were accurately known. The location of these couples and the distances between junctions are shown in the cross-section in Fig. 1. Since the method used was based on the determination of a uniform temperature gradient, the actual conductivity of the slab was not important and the effect of the thermocouple wires on the conductivity was of no consequence, provided that they did not cause irregularities in the temperature gradient.

The slab was poured in a frame in which the thermocouples had been stretched. This frame also contained two reference points across which a straight edge could be placed and the distances of the thermocouples were accurately determined with reference to the straight edge. In constructing the slab, the frame containing the thermocouples was placed on a plane surface with the two reference points up. The plaster of Paris was then poured without disturbing the thermocouples, and the upper surface was struck off with a straight edge drawn across the upper edges of the frame. After setting, this surface was sanded so that it presented a smooth plane face. By using a straight edge across the reference points, the position of the exposed slab surface relative to the thermocouples could be obtained at any time. The accuracy of the results depended to a large extent on the accuracy with which these distances could be determined.

An electric plate heater was placed against the other surface of the slab, with a thermocouple between the heater and the surface to serve as a control. This heater was insulated, so that most of the heat generated passed through the slab, and was provided with a rheostat and ammeter for adjusting and measuring the current. The electromotive force of the couples was measured by means of a precision potentiometer and all thermocouples were calibrated before installation.

TEST PROCEDURE

The three thermocouples used for indicating the surface temperature were installed and tested separately. The first thermocouple was made from No. 34 B. and S. gage copper and constantan wires with the junction filed to approximately the same diameter as the wires. It was laid on the surface, as shown in Fig. 1, and the junction and about 3 in. of the leads on each side of the junction were covered with a strip of thin adhesive vellum pressed tightly to the wire and the surface. When this series of tests was completed, the vellum was removed and a fine scratch was made in the surface. The junction and about 3 in. of the leads of this same thermocouple were then pressed into the scratch, sealed in with plaster of Paris, and worked flush with the surface. After this series of tests was completed, the same procedure was followed with a thermocouple made from No. 22 B. and S. gage copper and constantan wires. In this case, however, the junction and about 4 in. of the leads on each side were embedded to a depth of slightly more than one-half of the diameter of the wire, and, after the plaster had set, the protruding portion of the wire was filed flush with the surface. In all cases, the junction was placed on the axis of the slab and in line with the thermocouples located inside of the slab.

All tests were run with room air temperatures between the limits of 65 and 85 F. The temperature at the back of the slab, as indicated by the thermocouple placed between the plate heater and the slab surface, was brought to some predetermined value and was maintained constant until thermal equilibrium had been established. This required from one to four hours. When thermal equilibrium had been established several observations were made of the temperatures indicated by the thermocouples located inside the slab, on the surface, and in the air about 6 in. in front of the surface.

The observed temperatures as indicated by the thermocouples located inside

of the slab were then plotted on a graph in which the abscissae represented the location of the thermocouples relative to the surfaces of the slab, as shown in Fig. 2. Since the surface temperature desired was that of the undisturbed surface and not that produced and measured by the surface thermocouple, this temperature was represented by the intersection of the temperature gradient curve with the line *F* in Fig. 2 indicating the location of the exposed surface of the slab, provided that any disturbing influence caused by the presence of the surface thermocouple did not extend to the last thermocouple, *E*, located 0.163 in. below the surface. If the surface thermocouple truly indicates the surface temperature and causes no disturbance at the surface, then the reading of this thermocouple will be the same as that given by the intersection of the

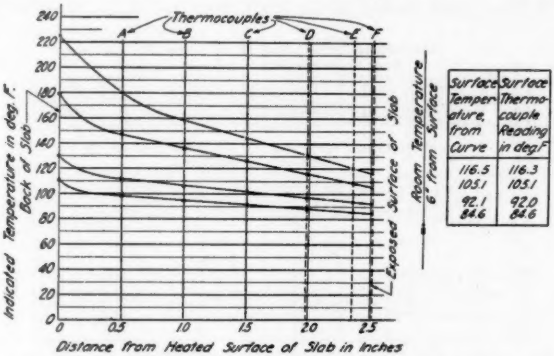


FIG. 2. TEMPERATURE GRADIENT IN PLASTER-OF-PARIS SLAB

lines on the graph. If on the other hand the surface thermocouple does not indicate the true surface temperature, or produces some disturbance tending to alter the true surface temperature, the difference between the temperature indicated by the surface thermocouple and that read from the intersection of the lines on the graph represents the correction to be applied to the reading of the surface thermocouple in order to obtain the true surface temperature. This procedure was followed in obtaining the corrections, and the correction was regarded as positive if the reading of the surface thermocouple was lower than the true surface temperature.

RESULTS OF TESTS

Sample temperature gradient curves are shown in Fig. 2, and the correction curves for the three types of surface thermocouples used are shown in Fig. 3. In the latter figure the corrections have been plotted against the difference in temperature between the exposed surface of the slab and the air, since it was not possible to maintain constant air temperatures for all tests. It was found that the actual air temperature had only a slight influence on the magnitude of the corrections over a range of about 20 deg.

It may be noted from Fig. 3 that for all practical purposes the thermocouple made from No. 34 B. and S. gage wire and embedded in the surface

indicated the true surface temperature. This same thermocouple placed on the surface and covered with thin vellum indicated a temperature lower than the true surface temperature. At a temperature difference of 40 deg between the surface and the air the correction was approximately +0.55 deg. The large couple made from No. 22 B. and S. gage wire and embedded in the surface also indicated a temperature lower than the true surface temperature, the correction amounting to +0.35 deg at a temperature difference of 40 deg between the surface and the air.

Since the large thermocouple was embedded rather deeply into the surface it seemed reasonable to expect that its reading would be influenced by the higher temperature layers just below the surface, and, if anything, that the indication of this couple would be too high, thus making a negative correction

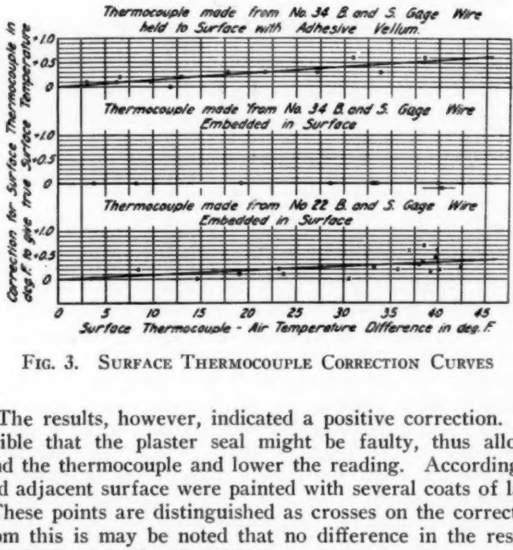


FIG. 3. SURFACE THERMOCOUPLE CORRECTION CURVES

necessary. The results, however, indicated a positive correction. It was considered possible that the plaster seal might be faulty, thus allowing air to diffuse behind the thermocouple and lower the reading. Accordingly, the thermocouple and adjacent surface were painted with several coats of lacquer to act as a seal. These points are distinguished as crosses on the correction curve in Fig. 3. From this is may be noted that no difference in the results obtained by the two methods of sealing is evident.

A reasonable explanation for the performance of the thermocouple made from No. 22 B. and S. gage wire may be based on the fact that, since the conductivity of the metal is relatively high, the heat loss by conduction through the metal wires would be greater than that through the same amount of plaster of Paris. Hence, the temperature of the wire and of the surface in the immediate vicinity of the thermocouple would be lowered over that of the remainder of the surface. The couple would indicate its own temperature correctly and would, therefore, indicate a value less than the true temperature of the undisturbed surface as obtained from the graph in Fig. 2, thus making a positive correction necessary, if the temperature disturbance did not extend in as far as couple E. The bright metal wires would tend to lose heat less rapidly by radiation than the surface, but the effect of the conduction loss

evidently more than offset this tendency to increase the surface temperature at the thermocouple.

In the case of the small thermocouple embedded in the surface, either the mass of the wire was not sufficient to lower the temperature of the surface adjacent to the thermocouple or the amount that it was lowered was just offset by the amount that the reading of the couple would tend to be increased by the higher temperature of the layers just below the surface. In the case of the small couple placed on the surface and covered with thin vellum, the presence of the vellum would tend to raise the temperature of the surface covered. However, the thermocouple and the vellum covering it extended out into the air which was at a considerably lower temperature than the surface,

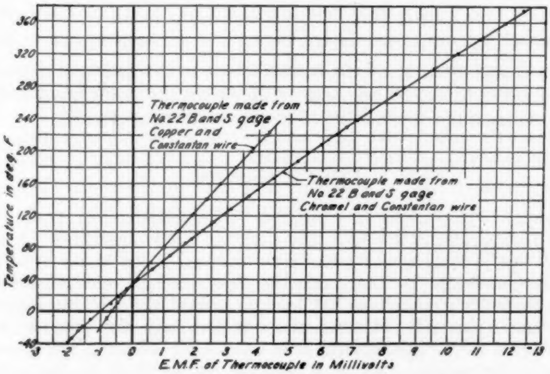


FIG. 4. THERMOCOUPLE CALIBRATION CURVES

and the net result was to cause the indication of the thermocouple to be lower than the true surface temperature.

For certain types of work it is sometimes advantageous to use thermocouples giving a higher electromotive force per degree temperature change than the commonly used, copper-constantan, iron-constantan, or chromel-alumel combinations. While the chromel-constantan combination is not novel, its use is rarely mentioned. Therefore, it seems worthy of some attention in connection with any study of thermocouples. In order to determine the characteristics of the chromel-constantan combination, three thermocouples were made from chromel and constantan wire and calibrated over a range of from −39 F to + 374 F. The calibration curve for one of these is shown in Fig. 4, together with the calibration curve for a copper-constantan couple to serve as a comparison. From these curves it is evident that the chromel-constantan couple is much more sensitive than the copper-constantan. For a given temperature the former generates an electromotive force approximately 50 per cent greater than the latter, and over the range tested no regions of discontinuity or inconsistency were found. Some trouble was experienced in forming the junctions, however, because these two metals are difficult to solder or fuse together.

The following conclusions may be drawn from the results of these tests as

applied to the use of thermocouples in connection with materials electrically insulating and having low thermal conductivity:

1. A thermocouple made of wire as fine as No. 34 B. and S. gage having the junction and 3 in. of the leads embedded in the surface and sealed flush with the surface with cement or plaster of Paris, for all practical purposes indicates the true temperature of the surface.

2. A thermocouple made of wire as large as No. 22 B. and S. gage having the junction and 4 in. of the leads embedded in the surface and sealed flush with the surface with cement or plaster of Paris indicates a temperature somewhat lower than the true temperature of the surface.

3. A thermocouple made of wire as fine as No. 34 B. and S. gage placed on the surface and having the junction and 3 in. of leads covered with thin vellum indicates a temperature lower than the true temperature of the surface.

4. Thermocouples made by the combination of chromel and constantan wire give electromotive forces 50 per cent greater than the electromotive force given by a copper-constantan thermocouple at the same temperature.

DISCUSSION

F. B. ROWLEY: This paper is a valuable contribution to the literature covering the use of thermocouples in the measurement of surface temperatures. The apparatus used and the test procedure was such as to leave no doubt in the relative accuracy of the different methods. I believe that the most common method used in the past, especially with building materials, has been to cement the wire to the surface with adhesive vellum. However, in some cases, it has been the practice to leave the couple exposed on the surface, holding it in contact by vellum or adhesive tape placed on each side of the couple. In Fig. 3, only the No. 34 B. & S. gage wire is shown as attached with vellum paper. However, it is evident that if the larger sized wire were used, the error would be still greater. Since it is usually practical to imbed the fine wire in the surface, it would be logical to adopt this method as standard practice when measuring surface temperatures of walls.

The calibration curves of Fig. 4 are of particular interest as it is often desirable to measure accurately differences in temperature. The chromel and constantan wire couples show a decided advantage for this work.

No. 944

SOME OBSERVATIONS ON HEATING PRACTICE

By James Govan,[1] Toronto, Ontario
(*NON-MEMBER*)

In a paper[2] presented by the writer at the Semi-Annual Meeting of the Society, 1929, information was submitted to show that the data used by the Society for determining maximum heating plant requirements for buildings should be modified. The purpose of the present paper is to submit the results of certain observations which have been made since that time and to offer certain suggestions with respect to assumed outdoor temperatures used for heat loss calculations in order to reconcile if possible the apparent discrepancy between actual and estimated heating plant performances.

Research conducted by the Society[3] has shown that a large error may be introduced into the calculations by failure to consider the periodic character of the heat flow through the walls and roof of a building. As has already been pointed out,[4] the resistance to heat flow and the heat capacity of the structure combine to stamp out this wave amplitude. The observations made by the author in Canada would tend to substantiate this fact, particularly with respect to the *fly-wheel* effect of the resistance to heat flow. In other words, materials of high heat resistance produce time-lag effects in both summer and winter air conditioning systems which must be taken into consideration in calculating the size of such plants.

The architectural profession is much concerned with this problem in its efforts to provide the most comfortable and healthful conditions for the occupants of buildings at the lowest possible capital and annual operating costs. If it is practicable, as seems possible from studies of actual occupied buildings under severe climatic conditions, to maintain comfortable conditions in a type of structure requiring as much as 70 per cent less heating plant than would be the case for the same building built in the customary manner, then it is folly to continue to follow present methods.

Unheated Packing Shed

Figs. 1, 2 and 3 show the plan and section and exterior and interior views of a packing shed in Ontario having frame wall and roof construction, with a

[1] Member of firm of Govan and Ferguson, Architects and Associated Engineers.

[2] Time Lag as a Factor in Heating Engineering Practice, by James Govan, A.S.H.V.E. Transactions, Vol. 35, 1929, p. 539.

[3] Coefficients of Heat Transfer as Measured Under Natural Weather Conditions, by F. C. Houghten and C. G. F. Zobel, A.S.H.V.E. Transactions, Vol. 34, 1928, p. 397.

[4] Heat Transmission as Influenced by Heat Capacity and Solar Radiation, by F. C. Houghten, J. L. Blackshaw, E. M. Pugh and Paul McDermott, A.S.H.V.E. Transactions, Vol. 38, 1932, p. 231.

Presented at the 39th Annual Meeting of the American Society of Heating and Ventilating Engineers, Cincinnati, Ohio, January, 1933.

fireproof board lining on the inside, fireproof insulation fill in stud and joist spaces, fireproof board, waterproof felt and clapboard on exterior of studs, and roof boarding and built-up roofing on top of roof joists. The windows consist of two single-glazed casement sashes bolted together with thumb bolts. Large doors at each end of the shed are of the overhead opening type and are single thickness; the small door in the side wall is also of single thickness,

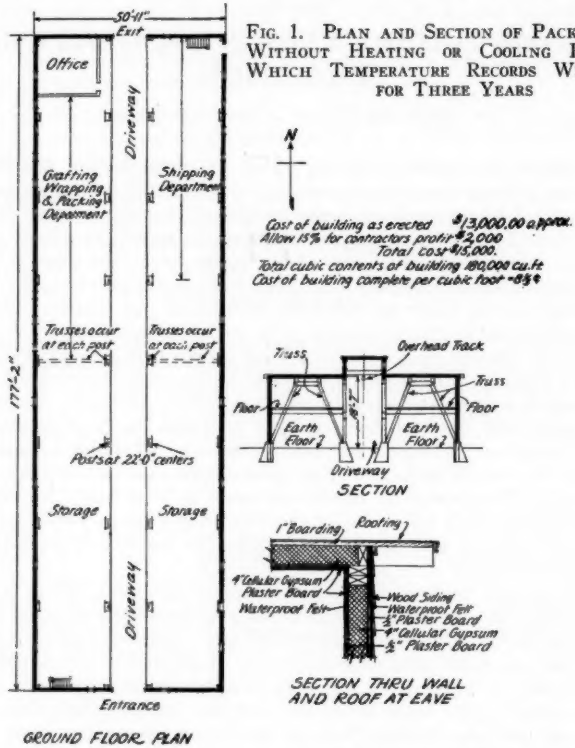


FIG. 1. PLAN AND SECTION OF PACKING SHED WITHOUT HEATING OR COOLING PLANT IN WHICH TEMPERATURE RECORDS WERE KEPT FOR THREE YEARS

and no caulking or weather-stripping was used at either windows or doors. The building was designed and built to provide a low-cost shed that would maintain a fairly uniform temperature during the nursery stock shipping seasons of the spring and fall. During the winter months it is used for grafting, budding and other indoor nursery occupations.

Indoor and outdoor temperature charts were taken from October 27, 1930, to February 23, 1931, and from March 23 to 29, 1931. The outdoor bulb of the recording thermometer was on the east and least exposed side of the building, which is built on an open rolling piece of land well exposed to the wind. Notwithstanding the fact that the outdoor temperature dropped to zero or lower on 14 occasions during this period, and to a degree or two above zero at other

times, the indoor temperature was as low as 30 F on only six occasions. The longest period of 30-F indoor temperature lasted only 24 hours and that followed a drop from 10 F to −17 F outdoors during the night of January 31-February 1, 1931. On January 28 it had dropped to −4 F, and during the night of the 30th-31st, to −4 F, rising to 10 F above at 3 p. m. on the 31st. The maximum variation indoors during that week was from 37 F to 30 F.

It is also worthy of note that, out of the 18 weeks, during 9 of them the outdoor temperature was consistently below the temperature indoors, and out of a ten-week period from November 24, 1930, to February 2, 1931, the outdoor temperature was higher than the indoor on only one occasion, namely,

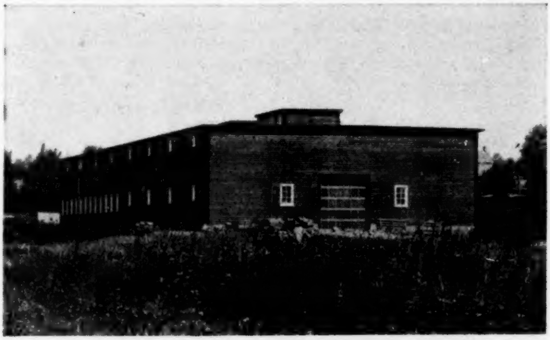


FIG. 2. EXTERIOR VIEW OF PACKING SHED

for about 8 hours on January 25, 1931, when the thermometer bulb hanging in the sun showed 40 F at about 2 p. m.

Suppose a heating plant were to be installed in this building to provide an inside temperature of 65 F for the men doing the winter grafting, etc. Disregarding the heat capacity of the structure, the outdoor temperature assumed would be −5 F or 15 deg below the lowest temperature on record, −20 F being the lowest recorded at the Nursery for 10 years. This would give a temperature difference of 70 deg when, as a matter of actual record, the inside temperature was maintained at 30 F or higher for two winters without any heating plant. If a temperature difference of only 35 deg were assumed (65−30), then the heating plant size would only be one-half that obtained by the usual methods of calculation. In other words, the plant ordinarily installed would be 100 per cent in excess of requirements.

The general formula used for calculating the amount of heating surface (radiation) required is as follows:

$$R = \frac{(H_v + H_i + H_t)\ (t - t_o)}{240} \tag{1}$$

where

R = square feet of equivalent heating surface (radiation) required, based on an emission of 240 Btu per square foot per hour

H_v = heat required to warm air introduced for ventilation purposes from t_o to t, Btu per hour

H_i = infiltration heat losses, Btu per hour

H_t = transmission heat losses, Btu per hour

t = inside design temperature, degrees Fahrenheit
t_o = outside design temperature, degrees Fahrenheit
240 = heat emitted by one square foot of equivalent heating surface (radiation) Btu per hour

Formula 1 may, of course, be used for either uninsulated or insulated buildings, provided the proper value of H_t is used for the type of construction involved. This formula has been modified by the writer for calculating the amount of heating surface required when the walls and roof have a comparatively high heat resistance, as follows:

$$R = \frac{(H_v + H_l + H_t)\ (t - t_x)}{240} \tag{2}$$

where

t_x = minimum indoor temperature that would be maintained indoors if no heating plant were provided (determined by observing the results in buildings after they are erected and modifying subsequent calculations accordingly)

SMALL OFFICE BUILDING

After the Semi-Annual Meeting 1929 an opportunity was sought to experiment with the small office building illustrated in Figs. 1, 1A and 2 of the paper presented at that meeting.[2] As then stated, the data regarding the heating surface in this office building had been checked by one of the then members of the A. S. H. V. E. Council, who satisfied himself that the amount of surface provided had not been over-estimated and was in accordance with the procedure set forth in THE GUIDE 1929.

Prior to the 1929-30 heating season, approximately one-third of each radiator was cut off and removed from the rooms. During that heating season it was found that the required temperature of the water at the boiler was still so low (in mild weather from 90 to 100 F) that the room temperature on the top floor farthest away from the boiler dropped below 70 F. To avoid the necessity of raising the temperature of the water at the boiler to stimulate circulation, the flow on each of the two main feed lines was increased slightly by mechanical means. This corrected the difficulty and the plant was operated in that manner during the 1930-31 season.

Fig. 4 shows the boiler with circulating devices on the mains at *A*, recording thermometer bulb in end of boiler at *B*, boiler water recording thermometer at *C*, and part of closed tank equipment at *D*. Indoor temperatures were recorded in the front room on the first and second floors and in the accounting room on the ground floor at the rear of building, also outdoors on the southeast wall, using a bare bulb not shielded from the sun and protected from the effect of north and west winds by the building. It should be noted that there is a Dominion Government Weather Recording Station on the same street not more than 200 yards from this building, and whereas the chart at the building shows a low dip to 2 F on January 8, 1931, the Government Record taken away from any building shows −2 F. Similarly other comparisons at these two points are as follows:

	Building Chart	*Government Chart*
January 15, 1931	1	−3
January 28, 1931	4	−1
February 1, 1931	−4	−6
February 3, 1931	0	−4
February 5, 1931	10	−3
February 11, 1931	1	−3

The temperature of 10 F shown at the building on February 5, 1931, contrasted with the Government record of −3 F may be explained by the retained heat in the wall affecting the nearby thermometer bulb, and also the heat coming from open windows in the accounting room which were quite close to the bulb. Frequently this room was occupied by staff employees working at night. This would be aggravated if the windows on the southeast side of this room were being used for ventilation with a wind from the northwest.

These charts show that from January 6, 1931, to February 11, 1931, the

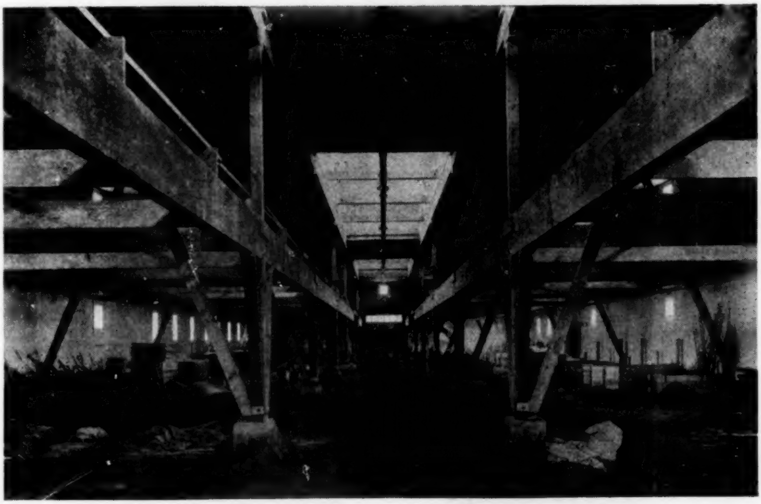


Fig. 3. Interior View of Packing Shed

indoor temperature dropped below 70 F on only one occasion, when it reached 60 F. The outdoor temperature dropped below zero at the Government Station, which would correspond to the north and west sides of the building, on eight different days.

Table 1 shows the temperature of the water at the boiler for the period during which the charts were made. On February 3, 1931, and February 5, 1931, it was below zero outside and yet the boiler water was not raised above 96 F until the 7th, and then only to 143 F for about one hour, which was the highest temperature recorded at the boiler during the heating season 1930-31. Without the reduction of 33 per cent in heating surface the temperature of water at the boiler recommended by the manufacturers of the pressure control tanks used is 190 F at an outside temperature of 0 F.

Coal Consumption

As the coal used was not weighed exactly as used each season, the only accurate figure than can be certified is that 75 tons of buckwheat anthracite were used for the whole of three heating seasons 1928-29, 1929-30 and 1930-31.

This gives an average consumption of 25 tons per season. As the building was only completed in the fall and early winter of 1928, the first season's coal consumption was naturally greater than that for the two following seasons, because of drying out of the building structure, etc. However, if 25 tons be taken as the yearly figure the facts regarding the poor combustion conditions at the boiler are interesting. It had been noted that due to the need for

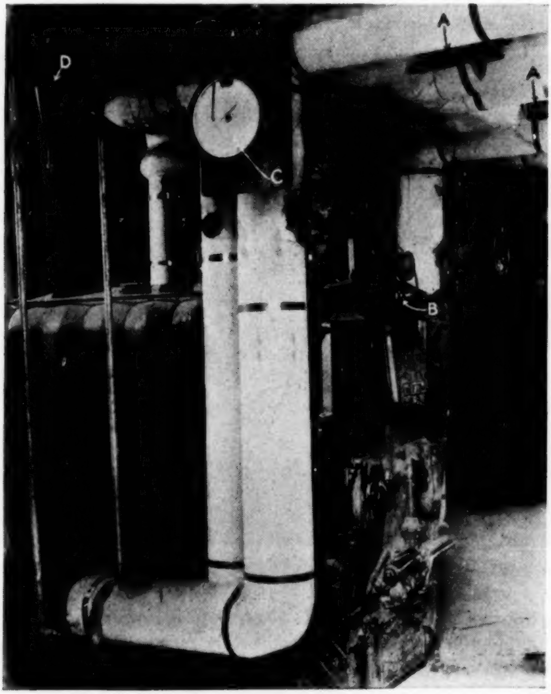


FIG. 4. BOILER IN SMALL OFFICE BUILDING

maintaining unusually low water temperatures at the boiler, it was practically impossible to maintain satisfactory fires, with the result that a very considerable proportion of the buckwheat sifted through the grates in an unburned condition. To check this the whole of the ashes for the season 1930-31 were kept in a pile and carefully sampled in the spring of 1931. This check gave over 20 per cent of the weight of ashes as unconsumed coal.

Further Reductions in Radiation for Season 1931-32

The studies made during 1930-31 suggested that the heating surface should be further reduced so that the boiler water temperature could be kept higher to maintain a better temperature head, and the grate area of the boiler reduced to obtain better combustion.

The total heating surface in use for the 1930-31 season was 1,296 sq ft. This was reduced by 448⅓ sq ft, bringing the total installed down to 847⅔ sq ft. Most of this reduction was made on the southeast and southwest sides of the building, because it had been noted that, notwithstanding previous reductions, the sun effect on these sides was quite pronounced. Fig. 5 shows the extent of the reductions in heating surface made since the original data were presented in the aforementioned paper.

The boiler is of the magazine-feed type with the gravity coal feed split two

Table 1. Temperatures of Water at Boiler (See Fig. 4)[a]

Date, 1931	High	Low	Mean	Date, 1931	High	Low	Mean
Jan. 6	131	100	119.3	Jan. 28	123	102	112.1
7	129	107	119.3	29	128	108	117.3
8	128	100	115.9	30	130	101	112.0
9	128	90	107.9	31	98	70	89.5
10	118	88	101.2	Feb. 1	97	75	90.3
11	123	105	110.5	2	...	...	....
12	119	99	101.0	3	95	78	87.5
13	131	103	120.0	4	94	74	87.4
14	119	103	109.6	5	96	79	83.7
15	133	109	119.2	6	96	74	88.1
16	129	106	119.8	7	143	90	119.8
17	120	90	103.6	8	131	92	119.2
18	128	86	110.2	9	128	98	116.5
19	132	104	120.5	10	130	104	112.6
20	134	110	125.0	11	126	100	117.2
21	134	109	125.1	12	120	105	113.0
22	134	110	123.5	13	130	104	115.1
23	135	110	124.6	14	141	94	118.5
24	128	95	112.6	15	133	95	113.5
25	130	90	112.9	16	118	92	108.2
26	129	105	115.5	17	122	95	107.5
27	132	110	121.7				

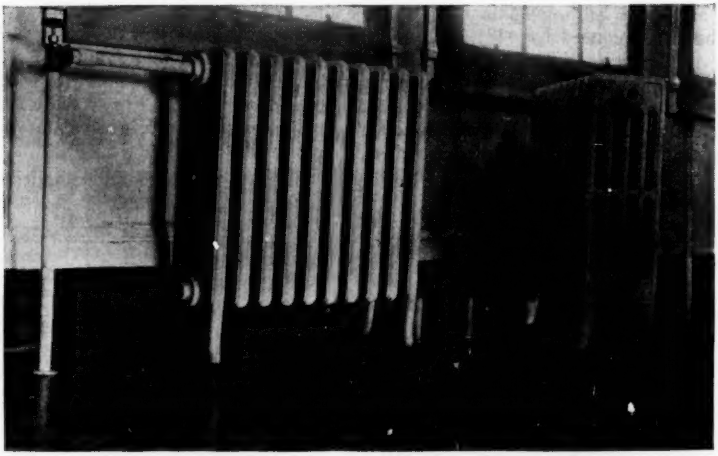
[a] The mean temperature at the boiler was 111.1 F for the whole period, January 6 to February 17.

ways, so it was decided that rather than cut down the length of each grate surface, one-half of the boiler should be shut down entirely. The boiler was operated in that condition for the whole of the 1931-32 season.

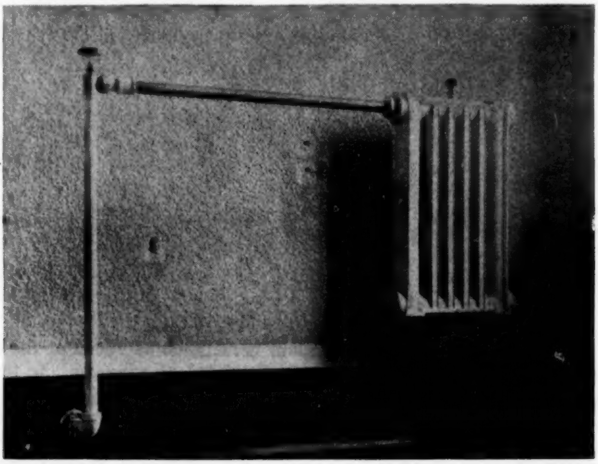
If the building in which the heating plant is installed had been of the ordinary uninsulated type of construction with the same kind of windows, the total heating surface required would be 2,830 sq ft. With only 847⅔ sq ft in use, the reduction is 1,982⅓ sq ft, which is approximately 70 per cent. An interesting fact is that the reduction in heating surface since this installation was previously reported to the Society in 1929 is approximately 50 per cent of the amount in use during the 1928-29 season, and as previously noted the plant at that time was designed in accordance with the procedure given in The Guide 1929, although the effect of heat capacity, discussed in The Guide 1931 and 1932, had not been taken into consideration.

Results During Season 1931-32

The reports obtained as to the comfort conditions in the building fully justify the reductions made in the fall of 1931. The weather was not as



Original installation, 26 sections or 91 sq ft; 7 sections or 24½ sq ft removed in 1929, leaving 66½ sq ft; 9 sections or 31½ sq ft removed in 1931, leaving 35 sq ft.



Original installation. 3 sections or 27 sq ft; 1 section or 9 sq ft removed in 1929, leaving 18 sq ft; 1 section or 9 sq ft removed in 1931, leaving 9 sq ft.

FIG. 5. TYPICAL REDUCTIONS MADE FROM ORIGINAL RADIATOR INSTALLATION [b]

[b] Operating results described in paper entitled Time Lag as a Factor in Heating Engineering Practice (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929). Cut made in summer 1929 removed one-third. Cut made prior to heating season 1931-32 shown by what is left connected up.

severe as in previous seasons, but the boiler water temperature records, in relation to the temperatures maintained in the building and the outdoor fluctua-

tions, prove that there is ample capacity in the plant to take care of much lower dips in outdoor temperature than were experienced during the 1931-32 season.

Temperatures were recorded for the week of March 9 to 16, 1932. The indoor temperatures were taken in the accounting room at the rear of building ground floor, and on the first and second floors. On both the ground and first floors the temperatures were higher than was desired for comfort, the

Table 2. Boiler Water Temperatures for Week of March 9, 1932

9th Day	10th Day	11th Day	12th Day	14th Day	15th Day	16th Day
170	165	132	145	135	135	139
156	155	133	143	132	132	135
142	142	134	140	130	126	128
130	132	132	130	146	118	115
154	131	158	135	144	126	127
150	146	164	137	143	124	138
145	147	161	137	141	123	135
138	132	150	139	135	121	131
138	139	160	137	136	130	130
144	130	159	136	131	135	123
130	124	140	122	131	134	130
120	117	133	112	136	131	134
123	115	131	106	137	129	135
128	115	131	110	135	127	134
135	115	132	120	132	124	131
141	115	131	125	133	120	129
144	116	134	129	130	118	127
147	116	130	130	128	116	125
149	116	128	130	126	115	124
147	115	124	126	124	114	122
145	113	120	123	126	113	121
160	125	133	...	122	132	120
168	130	148	...	138	139	120
170	131	152	...	130	141	129

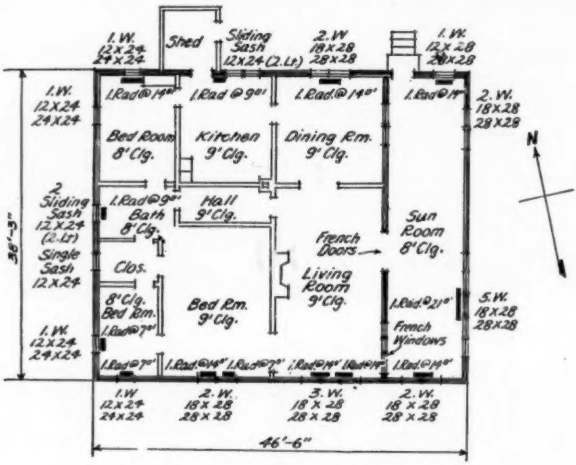
temperature on the ground floor being between 75 F and 80 F practically day and night, and on the first floor at 70 F or above.

These high temperatures were maintained because of conditions encountered in the large front room on the top floor. There the proportion of window to floor area is much larger than in any other part of the building. This room was originally planned for an assembly room for staff meetings, but has not been used for that purpose except on a few occasions. At other times it has been occupied by one or more individuals doing ordinary clerical work. This situation, coupled with the fact that the wood windows have no storm sash, and are in very loose condition, points to the need for increasing the relative heat retaining capacity of the room so that it will correspond with the other parts of the building. Alternatively, a small addition to the amount of heating surface in this one room would permit the maintenance of a lower water temperature at the boiler.

Table 2 gives the boiler water temperatures for this same week and shows a maximum of 170 F and a minimum of 106 F, with an average of 133 F for

the week. The average outdoor temperature for this period was about 25 F, the lowest being 4 F. According to the table of temperatures required for the closed system the boiler water temperature should have averaged 165 F for an average outdoor temperature of 25 F.

There were only two other occasions during the heating season when the boiler water temperature was over 170 F, *viz*, on March 7, 1932, when the



GROUND FLOOR PLAN

Total Heat Radiation
Installed—158 sq ft
4" insulation in walls shown blackened

3" insulation in floors
2 layers asphalt felt
4" insulation in ceiling

FIG. 6. PLAN OF BUNGALOW SUBJECTED TO TEMPERATURES BELOW −40 F

chart indicated 173 F for about half an hour and 176 F for about one hour on the following day. The outdoor temperature on the 6th and 7th did not go above the freezing point even with the bulb in the sun at noon, and on the 8th a temperature of 8 F was recorded at 6 a. m.

The only conclusion that an architect can draw from a study of this kind is that it would be much better to spend the small amount of money necessary to make one comparatively insignificant room in a job equal in heat retaining capacity to all the other rooms by improving the windows, rather than to spend 100 per cent more than is necessary on the heating plant to have a margin of safety that can never be put to use for the whole job.

BUNGALOW IN NORTHERN QUEBEC

Many other examples can be cited, but Fig. 6 showing the plan of a bungalow residence in Northern Quebec will give some idea of what is being done in Canada's Northland. This building is of frame construction and the walls, floors and ceiling are insulated with a fireproof insulation fill. The roof is not

insulated. No felt, caulking or other weatherstripping was used around storm sash or regular sash or doors.

Steam was provided for this residence and other buildings at a pressure of 25 lb at the plant boiler room located about 1,000 ft from the residence. The 25-lb pressure was the highest used all winter, although a temperature of −38 F was recorded during the 1929-30 winter. As higher outside temperatures were experienced the steam pressure was reduced to 15 lb.

The engineer reported that the super-heating effect of the steam being reduced from the boiler pressure of 80 lb to 25 lb at the valve located at the boiler was offset by the drop in temperature in the length of line. Condensation was noted in this main steam line at the residence. This was part of the evidence cited for the belief that super-heating effect was eliminated.

All heating supply and return pipes connecting radiators were in the space between the floor and grade. Supply pipes were covered with 3 ply cell covering. Return pipes were not covered. This amount of heat took care of the plumbing, sewers, etc., carried under floor. Radiators were not trapped individually, but one trap was provided on the supply steam and one trap on the return. The condensed steam was not returned to the boiler plant owing to local conditions.

The sun room was used continuously throughout the winter just the same as any other room. Double doors into it from other rooms were open continuously. This room was used for reading and sewing and as a child's play room, etc.

On January 4, 1930, when the temperature outdoors was −12 F, only 67 sq ft of heating surface was in use, as follows:

Bath Room	9 sq ft
Large Bed Room	7 sq ft
Living Room	14 sq ft
Sun Room	28 sq ft
Kitchen	9 sq ft
Total	67 sq ft

On that date the following conditions were noted: large bedroom window open and storm sash open 3 in.; living room fireplace flue open, no fire in fireplace for 3 days; kitchen outside door open into closed back porch; French doors and windows open between sun room and living room.

Kitchen, hallway, bathroom, dining room, living room and sun room were open through; bulbs in bloom and ferns in all main rooms. Under these conditions the living room temperature was 70 F, with an outside temperature at −12 F and only 67 sq ft of heating surface connected. Steam pressure at valve in boiler room was 20 lb.

Two Hospitals Compared

Two typical hospitals, No. 1 in Ontario constructed in a manner somewhat similar to the small office building described in this paper, and No. 2 in Quebec with ordinary construction, furnished some interesting comparisons. The ratio of occupants and staff in the Ontario and Quebec buildings is approximately 1 to 3, and the services and accommodation provided in the

Ontario building are, if anything, more elaborate than in Quebec, in proportion to the number of patients accommodated.

	Ontario Hospital No. 1	*Quebec Hospital No.* 2
Cubic space heated	160,000 cu ft	461,000 cu ft
Coal consumption per year	60 tons (Heating), 14 tons (Domestic water)	800 tons
Electric light and power cost per year	$785.65	$3,516.00
Gas cost per year for cooking	None	900.00

These two institutions are located in districts where the climatic conditions are similar and there is nothing radically different about them except the building construction and the mechanical and heating equipment. Sterilizing and other provisions affecting comparison just as complete in No. 1 as in No. 2. Both winter and summer comfort conditions for patients are better in No. 1 than in No. 2 and the capital cost per patient was at least 30 per cent lower in No. 1. Hospital No. 2 is not an abnormal example of such institutions. In fact, it may be said to be typical in both first and maintenance costs of the average institution of this kind built all over Canada and the northern section of the United States. The cost per 1,000 cu ft for heat, light and power compares very closely with several other typical institutions studied.

SUMMARY AND CONCLUSIONS

1. Under the extreme variations of temperature in Canada and the Northern States, the most important factor in determining heating plant size appears to be the outdoor temperature assumed, particularly when well-insulated types of construction are used.

2. Estimates of annual fuel consumption based on outdoor mean temperatures obtained from instruments shielded from the sun (as is the case at most of the Canadian Government Weather Stations) are likely to be seriously in error.

3. More attention should be paid to balancing the heat retaining capacity in all parts of a building, so as to avoid the necessity of providing a large, uneconomical reserve in heating plant capacity for only rare emergency use.

4. The heat radiating from the uncovered earth floor of the building in winter and absorbed by the earth in summer plays an important part in stabilizing the indoor temperature in the buildings studied by the writer. For instance, in the large storage shed illustrated in Figs. 1, 2 and 3, the earth temperature drops to about 40 F at a point 6 in. below the surface at the end of the winter.

5. The fact that an indoor temperature of 30 F or higher has been maintained in buildings without any artificial heating plants for three consecutive winters, where the outdoor temperature has dropped to −20 F, and also that 70 F has been recorded indoors without artificial cooling when the outdoor temperature was 90 F, indicates the need for further investigation to provide reliable data on the fly-wheel effect of different types of construction.

No. 945

AIR SUPPLY AND ITS EFFECT ON PERFORMANCE OF OIL BURNERS AND HEATING BOILERS

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and E. J. Tavanlar [3] (*MEMBER*)
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This paper is the result of research sponsored by the American Society of Heating and Ventilating Engineers in cooperation with the *American Oil Burner Association* and conducted at Yale University

THIS paper deals with certain performance features related to the manner in which the air for the combustion of oil is supplied. It proposes to show (1) the effect of furnace draft variations on combustion, boiler efficiency, flue temperature and draft loss through the boiler, (2) a probable cause of draft difficulty in practice, (3) limiting factors relative to capacities obtainable, and (4) a possible explanation for inability in some cases to obtain the full rated capacity of a burner. Tests were made on four oil burners. The essential distinctions between these burners are given in Table 1.

Effect of Furnace Draft Variations

The burners were set for a fuel rate averaging about 16½ lb per hour; the furnace draft fixed at 0.02 in. of water and the air regulated to produce a carbon dioxide (CO_2) content of 10 per cent or about 50 per cent of excess air. The efficiencies under these conditions which, although not all the same, were consistent as to type.[4]

The furnace draft was next increased to 0.07 in. of water. Fig. 1 shows that the carbon dioxide (CO_2) and efficiency decreased while the flue temperature and total draft increased, the degree of change being somewhat variable as regards types of burners. This is substantiated by previous tests of a similar character.[4]

If the draft loss through the boiler is calculated (total or smoke-pipe draft—furnace draft) it shows that this value rises rapidly as the carbon dioxide (CO_2) decreases (Fig. 2). It will be noted that seemingly little or no draft loss exists through the boiler at 0.02 in. of furnace draft. This is not strictly

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[3] Graduate student in Mechanical Engineering, Sheffield Scientific School, Yale University.
[4] For additional data see A. S. H. V. E. research paper entitled, Study of Performance Characteristics of Oil Burners and Low-Pressure Heating Boilers, by L. E. Seeley and E. J. Tavanlar, A. S. H. V. E. Transactions, Vol. 37, 1931, p. 517.

Presented at the 39th Annual Meeting of the American Society of Heating and Ventilating Engineers, Cincinnati, Ohio, January, 1933.

TABLE 1. ESSENTIAL FEATURES OF FOUR TYPES OF OIL BURNERS TESTED

Burner	Method of Air Supply and Regulation	Oil Regulation
A	Practically entire supply by natural draft-control by furnace draft and slide damper in opening below furnace. NOTE: In practice slide damper would be adjusted and remain fixed thereafter.	Independent though mixed with small amount of air in pump under pressure to assist atomization.
B	Primary air supplied by fan, entering furnace with the oil in manner intended to achieve good mixing. Secondary air induced by natural draft through opening to furnace located below burner. Primary air control by fan adjustment and furnace draft. Secondary air control by size of opening and furnace draft.	Adjustment in feed line though fuel and primary air adjustments probably interdependent with regard to insuring good mixing.
C	Similar to *B* except in degree—primary air supply relatively larger proportion of the total. Control similar.	Oil feed solely dependent upon rate of primary air feed although controllable initially to some extent at the time of installation.
D	Entire supply from single source under comparatively high pressure. Controlled by suitable fan adjustments and to a slight extent by furnace draft.	Independent regulation.

true, but the loss in the boiler used was very small. It must also be remembered that whenever a vertical flue passage exists there is a chimney effect present within the boiler itself which assists in the movement of the gases. A special gas boiler with a non-mechanical burner is an excellent example where the chimney itself is of no direct help to this boiler because of the hood or back-draft diverter employed. Small, vertical, coal-fired boilers at certain outputs will show a draft reading in the smoke-pipe less than that in the furnace itself. Regardless of this, the smoke-pipe draft is the one that must be produced by the chimney at the boiler outlet and Fig. 2 shows that this requirement does increase rapidly with a decrease in carbon dioxide (CO_2).

POOR CHIMNEY DRAFTS

One explanation for chimney draft troubles which has not received the attention it deserves may be found from the foregoing statement. It is generally understood that a low percentage of carbon dioxide (CO_2) indicates a large amount of excess air. The actual relation between excess air and per cent carbon dixode (CO_2) is not so commonly understood (see Fig. 3) and more particularly the principles that relate to the flow of gases.

The latter may be explained by considering the following facts:

1. All gases resulting from combustion, as well as excess air and nitrogen, must flow through the boiler flues, the breeching or smoke-pipe and the chimney.
2. All the flow areas of these passages are fixed in amount.
3. The velocity of the gases through these respective parts, therefore, depends upon the volume of gases to be handled.

4. A certain pressure difference or draft will be required to achieve and maintain these velocities against the frictional resistance of flow.

5. The velocity varies as the square root of the draft head (Velocity = Constant $\sqrt{h}$).

It is evident, therefore, that if twice the volume of gases results in one case as compared to another, due to different air adjustments, the draft required

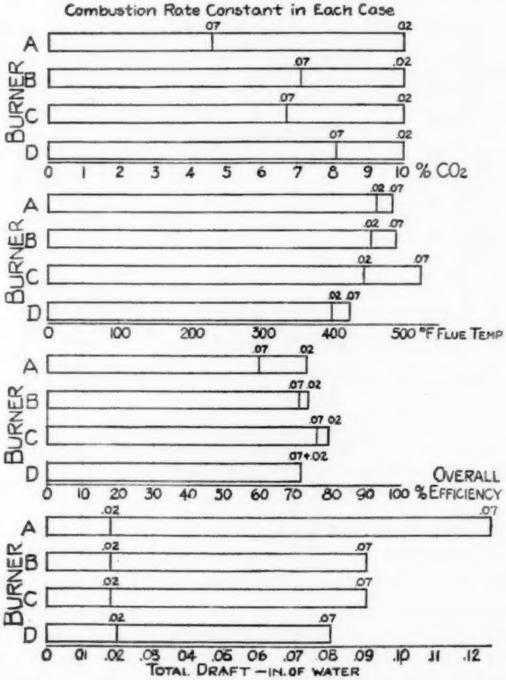

Fig. 1. Effect of Change in Furnace Draft from 0.02 to 0.07 in. of Water

will be four times as great, even though the velocity is only doubled. Whereas this example may represent the difference between a good and a poor installation, the variation in draft requirement is so great that in practice the difference between a *moderately good* and an *only fair* adjustment may be critical if the chimney happens to be *weak*.

The difficulties become cumulative. A poor adjustment calls for a higher oil feed due to reduced efficiencies, thus increasing the volume of flue gases. Furthermore, practically all of the tests to date have shown that an increase in excess air increases the flue gas temperatures. This merely aggravates conditions because the effect is to increase the volume of flue gases beyond that due to the excess air alone. Although this increased temperature may

improve chimney draft to some extent, any resulting increment in draft will always be insufficient to produce the increment in velocity which the same event makes necessary.

Furthermore, oil burners, with few exceptions, are set to produce the maximum heating load imposed by the heating system. This means that the

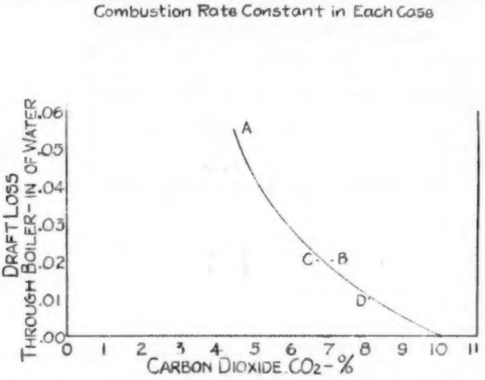

FIG. 2. RELATION OF DRAFT LOSS THROUGH BOILER TO PER CENT CO_2

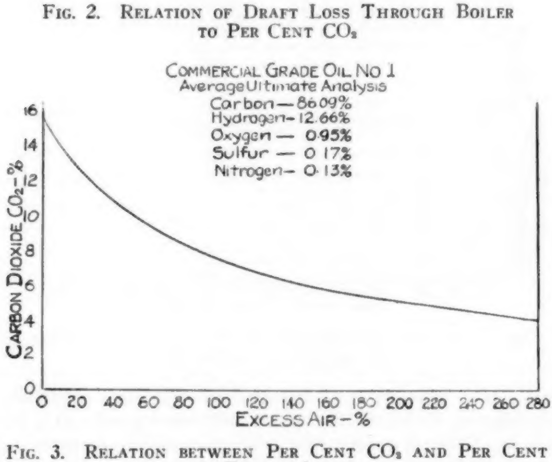

FIG. 3. RELATION BETWEEN PER CENT CO_2 AND PER CENT EXCESS AIR

chimney is expected at all times to handle comparatively large quantities of gases. This is another reason for not adding the *final straw* which in this case is excess air. It is not to be assumed from the foregoing that a great many chimneys are inadequate for oil burners. There are *border line* chimneys, however, that would prove troublesome for any method of fuel burning and it is with these that care is required. If a draft condition is bad and no

other improvement can be made, the excess air should be reduced if possible—perhaps the rate of oil feed also in extreme cases.

It is evident if oil and air adjustments are to be well made and maintained that control of furnace draft is absolutely vital to burner *A*, important as to *B* and *C* and comparatively unimportant to *D* (Fig. 1).

EFFECT ON OVERALL EFFICIENCY

The efficiency curves based on intermittent operation give the truest picture of the average efficiencies obtainable in practice. In order to study this feature of the work the curves were plotted by arbitrarily shifting them from

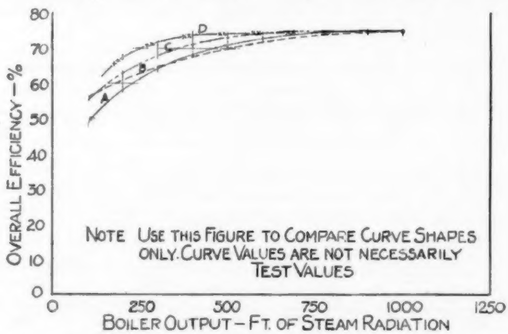

FIG. 4. INTERMITTENT EFFICIENCY CURVES, SERIES E
0.02 in. of water furnace draft, 10 per cent CO_2 during on period, 1,000 ft—radiation boiler output

their true values to coincide at 75 per cent efficiency and 1,000 ft-*radiation* boiler output, (Fig. 4). This was done solely for the purpose of comparing the curve shapes and the graph should not be used for any other purpose.

It shows quite clearly that the order of preference would be *D, C, B* and *A* if during operation they all produced exactly the same efficiency and output as shown by Fig. 4. The differences reflect the relative ease with which air can continue to flow into and through the boiler during the *off* period and thus carry away heat previously stored in the boiler and refractories.

When this is coupled with the fact that the curves could all be lowered by an increase of furnace draft an amount equal to the decrease in efficiency shown in Fig. 1, the influence of the method of air admission becomes most striking.

FACTORS INFLUENCING CAPACITY LIMITS OF OIL BURNERS

The following factors may act as limits on the capacity or fuel-burning rate of an oil burner.

General Considerations

1. The heating load must in practice be the determining factor although there is a lack of uniformity in the allowances made to establish the total or

gross heat output required. This should be corrected because the basic problem is one of heat loss from building structures—a problem essentially common to all methods of heat generating. This does not imply equal allowances in all cases but a determination by means of common method of approach.

2. The efficient heat absorbing rate of the boiler or equivalent device should be the most desirable limiting factor. Proper matching of Items 1 and 2 would secure high standards of performance.

Specific Considerations Relating to Oil Burner

3. *The furnace* size and design in conjunction with type of burner and effectiveness of mixing may impose a limit on fuel burning or heat release

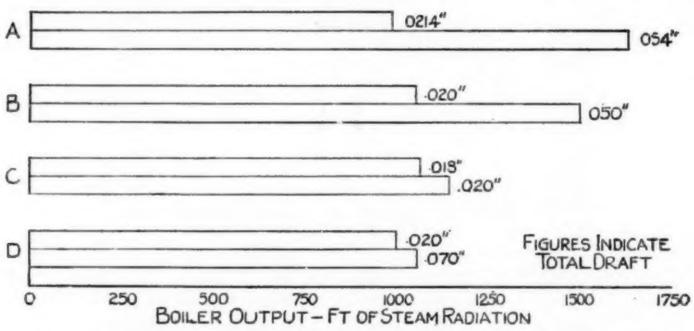

FIG. 5. RELATION OF TOTAL DRAFT TO LIMITS OF MAXIMUM BOILER OUTPUT—10.0 PER CENT CO_2 IN ALL CASES

rate. There is no standard value, however—where 35,000 Btu per cubic foot of net furnace volume per hour seems reasonable for one burner, 80,000 Btu may be reasonable for another. Rates of 180,000 Btu have been obtained with no apparent difficulty. It cannot be concluded that the limits have been satisfactorily explored.

4. The *oil supply* may be limited by pump capacity, size of orifice and fuel pressure, by primary air supply when the method of oil feed is dependent thereon (*i.e.* carburetor principle and oil burner type *B*) or by the total air supply no matter how obtained.

5. The *air supply* may be limited by fan capacity, size of air-supply openings and also by available furnace draft.

Fig. 5 was used as the basis for estimating the probable limiting factors for the burners tested. The values were obtained as follows:

1. The quality of combustion (*i.e.* per cent carbon dioxide) was required always to be at 10 per cent and the flame clean.

2. The air and oil supply and draft was increased, if possible, with the above restriction.

3. The boiler output was measured in every case.

Limits for Burner A—Oil and draft (or air) were not limited—the limit was in the furnace, probably in mixing. If a larger proportion of excess air had been allowed the fuel burning rate might have been increased.

Limits for Burner B—The limit here is problematical—it may have been an atomizing, a primary air or a furnace limitation. It was not secondary air because this could have been increased by raising the draft.

Limits for Burner C—Primary air limit undoubtedly; more primary air would have produced more oil; secondary air could have been considerably increased. Except at maximum capacity no secondary air is needed and only comparatively little if 10 per cent carbon dioxide (CO_2) is to be maintained. Probably better to dispense with secondary air in this case and avoid its disadvantages.

Limits for Burner D—Air capacity limit—increase in draft failed to add much air. It should be pointed out that a secondary air supply might be devised, making it possible to increase the oil supply and produce the characteristics of Burner *B*.

If it had been the express purpose of this investigation to analyze the precise limits of an intermediate type such as Burner *B*, the oil and air could be increased and if it became necessary to change their relative proportions (*i.e.* increase the excess air) in order to maintain a clean flame, it would clearly indicate a furnace limit of some sort—probably mixing.

Mixing depends upon fuel preparation (*i.e.* degree of atomization, completeness of vaporization and temperature and location of refractory surfaces), velocity of air supplies which includes direction as well as shape and size of furnace. The complications become self-evident. So many solutions actually exist that some burners are found highly independent of and some very dependent upon furnace design in order to secure proper mixing and complete combustion. This, in a measure, explains why the rate of heat release per cubic foot of furnace volume varies so widely in different installations.

Relation of Excess Air to Rated Capacity

Fig. 3 indicates one reason why an oil burner may be considered to be too small when its rated capacity indicates that this should not be the case. If, for example, a burner supplying all its air by means of a fan, is adjusted to 12½ per cent carbon dioxide (CO_2) with a certain furnace draft, the maximum rate of fuel burning will be established when the fan is wide open. If the burner is rated at this value or even somewhat less and then placed in the field where an adjustment of only 4.8 per cent of carbon dioxide (CO_2) is obtained—a possible event—it means that twice as many pounds of air per pound of fuel are being supplied. The result is that only about one-half of the original amount of oil can be burned. If the furnace draft is the same the fan can deliver no more air on the job than in the laboratory. The boiler output will be reduced usually more than half due to lower efficiencies. The burner is considered to be too small when a better adjustment of oil and air (*i.e.* decrease in excess air) might remedy the trouble. The adjustment of oil and air cannot be properly measured except by analyzing the gases by means of an Orsat.

While the foregoing is not the sole cause for such a condition, (*i.e.* the furnace design may be unsatisfactory) it is still true that the actual quality of performance conclusively rests upon those who install the equipment.

CONCLUSIONS

1. In a burner which receives its air supply by natural draft exclusively the quality of combustion is very sensitive to draft fluctuations. This is important because of the adverse effect on efficiency. In practice the supply of air could be either too great or insufficient unless carefully adjusted and precisely regulated thereafter. During intermittent operation its stand-by or off-period losses will run higher than other burners.

2. A burner which receives its air supply exclusively from a fan will generally be the least sensitive to draft changes and more satisfactory during intermittent operation. This will differ in degree depending upon the relative resistance which the fan and other parts set up to the flow of air during the off-period.

3. Intermediate types will generally occupy the middle ground between the preceding two. There will never be a sharp line of demarcation and a certain amount of overlapping is to be expected.

4. If a means could be found to stop the flow of air through a boiler during the off-period regardless of type of burner the foregoing differences on intermittent operation would virtually disappear and the average efficiency would be increased.

5. Where unsatisfactory draft conditions are encountered, excess air should be reduced as much as practicable, the furnace draft should be kept reasonably low and provision should be made for automatic draft regulation.

6. If the maximum fuel burning rate proves to be insufficient for the job it may in some cases be raised by reducing the excess air. Furnace alterations to secure better mixing and combustion may also be necessary.

7. The greatest improvements and the hope of maximum success lie in the field of installation.

No. 946

COLD WALLS AND THEIR RELATION TO THE FEELING OF WARMTH

By F. C. Houghten * (*MEMBER*) and Paul McDermott † (*NON-MEMBER*)
Pittsburgh, Pa.

THE effects which the thermal properties of man's atmospheric environment have on his feeling of warmth and his body reactions have been the subject of extensive research carried on in recent years by the physiologist and the air-conditioning engineer. Since 1920 a series of investigations of this subject have been carried on by the A. S. H. V. E. Research Laboratory in cooperation with the United States Bureau of Mines and the United States Public Health Service. Results of the investigations have been published in the Transactions of the Society. Similar studies have been made by Harvard University.

Thermal equilibrium in the human organism is maintained largely through dissipation of heat to the surrounding atmosphere through evaporation of perspiration and through convection, and by radiation to surrounding surfaces. Evaporation of perspiration, either sensible or insensible, depends upon the temperature and vapor pressure of the atmosphere and the availability of moisture on the body. Air motion is necessarily an important factor in heat loss by evaporation, since it removes the moisture-laden air near the body and replaces it with air having greater capacity for removing moisture. As demonstrated by an earlier Laboratory study,[1,2] the control of the amount of perspiration available for evaporation is the body's most responsive method of regulating heat dissipation; through its reflex control of perspiration, the body maintains temperature equilibrium over a wide range of atmospheric conditions and for varying rates of heat production resulting from increased activity.

Heat dissipation from the body to the surrounding air through contact and by convection currents is generally referred to as convection loss, and depends

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1 Thermal Exchanges Between the Human Body and Its Atmospheric Environment, by F. C. Houghten, W. W. Teague, W. E. Miller, and W. P. Yant, *American Journal of Physiology*, Vol. 88, No. 3, April, 1929.

2 Heat and Moisture Losses from the Human Body and Their Relation to Air Conditioning Problems, by F. C. Houghten, W. W. Teague, W. E. Miller, and W. P. Yant, A.S.H.V.E. Transactions, Vol. 35, 1929, p. 245.

Presented at the 39th Annual Meeting of the American Society of Heating and Ventilating Engineers, Cincinnati, Ohio, January, 1933, by F. C. Houghten.

upon the temperature difference between the body surface and the dry-bulb temperature of the air. Air movement over the surface or through the clothing removes the warm air next to the body surface and replaces it with cooler air, thus also having an important effect upon the heat loss by convection.

Heat loss by radiation takes place directly from the outside visible surface of the clothed body to the visible surfaces of all surrounding walls and objects in view of it. Although the heat lost by radiation passes through the intervening air, it is for all practical purposes independent of the physical properties of the atmosphere. Such heat loss is in accordance with the Stefan-Boltzmann radiation law and depends upon the surface temperature and emissivity of the visible areas of the clothed body, the solid angle subtended at a particular point on the body surface by the surfaces of the walls and other objects, and on the temperature and emissivity of the surface of the walls and other objects in view. Because air circulates through and under the clothing which normally covers a large part of the body, the temperature of the visible surface of the clothed body is more nearly that of the air than that of the body surface or 99 F. Hence, the clothed part of the body is much less subject to variation in heat loss by radiation than is the smaller area of the uncovered portion of the body surface. Within the limits met in practice, the varying surface characteristics of clothing, walls, etc., have little effect on heat dissipation by radiation because, at temperatures commonly found, the emissivity constant for such surfaces is assumed to have a maximum value or unity.

Most of the early studies by the Laboratory and cooperating organizations covering the effect of atmospheric environment on man's feeling of warmth were made in rooms surrounded by well insulated interior walls. There were two objects in doing this: first, to simulate conditions in buildings to which air conditioning based upon the findings was most likely to be applied; and second, to simplify an already complicated problem by the elimination of the variable radiation factor. Although there was no misunderstanding of the importance which it plays in the dissipation of body heat, there may have been some under-estimation of the degree to which heat loss by radiation may vary under different types of building construction and occupancy. The two factors which minimize the variable effect which radiation plays in heat loss from the body are the effect of clothing and the comparatively small differences in the respective inside surface temperatures had by walls of common types of construction when exposed to the same outside and inside air temperatures. The temperatures of inside surfaces of exterior walls of air conditioned space follow more closely the inside air temperature than they do the outside temperature. Three walls built-up as follows will give about as wide a variation in thermal transmission as is met with in building practice: first, a very cheap and poorly insulated construction, made of inside plaster on lath, studding, and outside siding; second, an 8-in. brick wall with 1⅝-in. furred space, lath and plaster; and third, a 16-in. brick wall with 2-in. cork insulation and plaster. These three walls have air to air transmission coefficients of 0.395, 0.250, and 0.094, respectively. If an inside film conductance of 1.6 is assumed, under an outside temperature of zero and an inside temperature of 70 F, the resulting inside wall surface temperatures will be 52.7 F, 59.1 F, and 65.9 F, respectively.

The importance which variation in radiation may play in maintaining thermal equilibrium of the body may be calculated if the magnitude of the factors in-

volved are known. According to Stefan-Boltzmann's law, heat dissipation by radiation is given by the equation:

$$R = 0.172 \times 10^{-8} A \epsilon (T_1^4 - T_2^4) \tag{1}$$

where

R = the heat dissipated by radiation in Btu per hour.
ϵ = the emissivity of the surfaces or unity for both the surfaces of the clothed body and the walls for low temperature radiations.
A = the area of the human body effective in radiating heat, assumed to be 17 sq ft.
T_1 = the absolute temperature of the body surface assumed to be equivalent to an average of 75 F for the visible surface of the clothed body.
T_2 = the absolute temperature of the surrounding wall surfaces in view of the body.

Substituting these values in Equation 1 and assuming that the entire surrounding wall surface or a surface subtending a solid angle of 4π steradians is at air temperature of 70 F, a rate of heat dissipation by radiation of 101.4 Btu per hour is given. If all values remain the same as assumed, except that one sixth of the solid angle at the body is subtended by surfaces at 60 F instead of 70 F, the radiation loss will be 133.4 Btu per hour, or 32 per cent greater than in the warm wall room. While the value of the factors entering into the problem and therefore the numerical results cannot be offered as applying rigidly, they do serve to demonstrate the extent to which a change of 10 deg in only a part of the surrounding walls affects that part of the total heat loss which is dissipated by radiation, emphasizing the fact that the difference between the fourth powers of two numbers is much greater than that of the numbers themselves.

Since the feeling of comfort as regards warmth for a person seated at rest depends upon the balance between his rate of heat production and heat loss, any increase in heat dissipation by radiation must, to maintain the same feeling of warmth, be compensated for either by increased metabolism or by a corresponding decrease in the combined loss by evaporation and convection. If the metabolism remains constant, the required decrease in evaporation and convection may be brought about by increasing the air temperature. The purpose of the present study was to evaluate the cooling effect on a person because of excessive radiation of heat to the surfaces of walls at temperatures lower than the room air, and to determine the higher air temperature necessary to compensate for this excessive radiation loss.

TEST ARRANGEMENTS AND PROCEDURE

The study reported in this paper was made in the psychrometric chambers of the A. S. H. V. E. Research Laboratory located in the Pittsburgh Experiment Station of the United States Bureau of Mines. These chambers and the equipment for air conditioning them are described in other Laboratory reports.[3] The chambers as originally built and in which all the earlier Laboratory comfort studies were made are entirely surrounded by interior walls of heavy construction with 4-in. cork insulation. The exceedingly low conductances of these walls maintains their inside surface temperatures practically equal to the tem-

[3] Determining Lines of Equal Comfort, by F. C. Houghten and C. P. Yaglou, A.S.H.V.E. TRANSACTIONS, Vol. 29, 1923, p. 361.

perature of the air in the chamber after constant temperature conditions have been established.

For the study of the effect of radiation, a 5 ft by 6 ft test room, Fig. 1, whose wall temperatures could be controlled independently of the air temperature, was built in the second psychrometric chamber, with a door between it and the first psychrometric chamber, so that a person could pass quickly from the small test room into the first psychrometric chamber. These rooms are referred to as

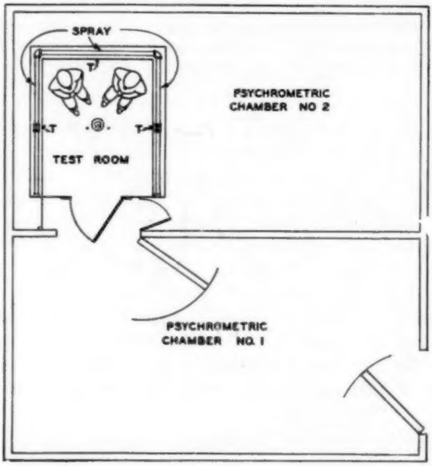


Fig. 1. Cold Wall Test Room in Psychrometric Chamber No. 2

the test room and the control room, respectively. The walls of the test room were made of sheet-metal, painted on the inside with a light buff pigment paint.

A vertical cross section of the test room which shows the auxiliary equipment for maintaining constant wall temperatures independently of the air temperature is shown in Fig. 2. Three of its sheet-metal walls were cooled by cold water sprays, but the fourth, containing the door, was not. Thermostatically controlled water was sprayed on the tops of the outside surfaces and made to flow in a uniform sheet down the entire area of the three sides of the room in sufficient volume to hold the inside surface temperature of the walls constant for the same elevation and to keep the difference in temperature between bottom and top within 0.5 deg. Thermocouple junctions for temperature measurements were attached to the center, at the top, and at the bottom on the inside surfaces of each of the three cold walls.

Air was admitted to the test room through the lattice work ceiling and left it through the lattice work floor. Screen baffles and resistances were so placed above the ceiling and below the floor that a uniformly distributed air flow was had throughout the room. Air was supplied from the air conditioning apparatus at desired temperatures and humidities in volumes that would give a suitable balance between velocity effect on the subjects and the fall in air

temperature from ceiling to floor due to cold walls. A volume of 31,800 cu ft per hour was circulated, which for the unoccupied room with 50 F walls and 80 F entering air gave an air temperature drop of 5.5 deg from ceiling to floor. With other conditions the same, two subjects in the room reduced this temperature drop by approximately 0.7 deg. The calculated linear velocity through the cross section of the room necessary for 31,800 cu ft per hour of air passing was 26.5 fpm; however, the Kata thermometer reading gave

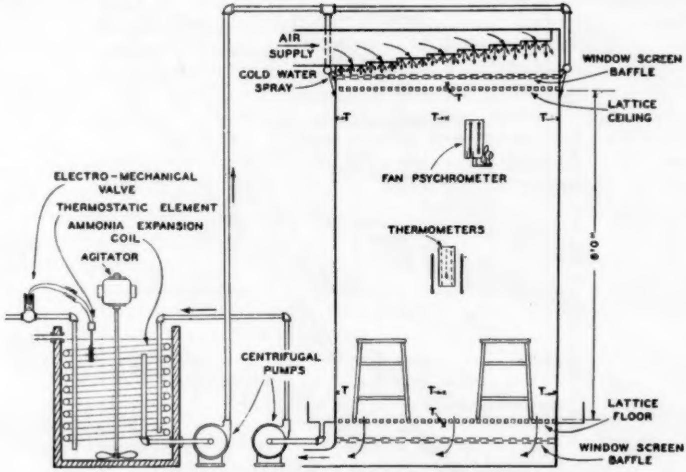


Fig. 2. Cold Wall Test Room and Control Equipment

an air velocity of 33.4 fpm. The difference between this velocity and 26.5 fpm as calculated should be accounted for by eddy currents. The Kata reading in the control room gave velocities ranging from 20 to 30 fpm.

Air temperatures at the 30-in level in the center of the rooms were determined, both with and without occupants in the room, by a plain mercury thermometer and a shielded mercury thermometer. Temperatures were taken at the ceiling and the floor with shielded thermocouples, and at the 5-ft level the wet and dry-bulb temperatures were observed with a fan psychrometer. For any given setting of the air conditioning equipment, the wet and dry-bulb temperatures were observed with no occupants in the room, and the resulting moisture content applied to the changed condition in calculating the relative humidity of the occupied room at the 30-in. level. In order to avoid the effect of excessive air motion in close proximity to the subjects, the fan on the psychrometer was not run during occupancy.

The study, made from February to July, includes both the summer and winter seasons, but since the subjects were exposed to approximately the test conditions for at least two hours before observations were made, outside weather conditions should have little effect on the results. Three subjects participated in the study; however, because of the small volume of the test room, only two

were allowed in it at a time. The tests were made with cold wall temperatures in the test room of 45 F, 50 F, and 60 F, with dry-bulb temperatures in the control room ranging up to approximately 80 F, and at varying relative humidities ranging from 20 to 80 per cent.

The individual tests on subjects were conducted in a manner similar to

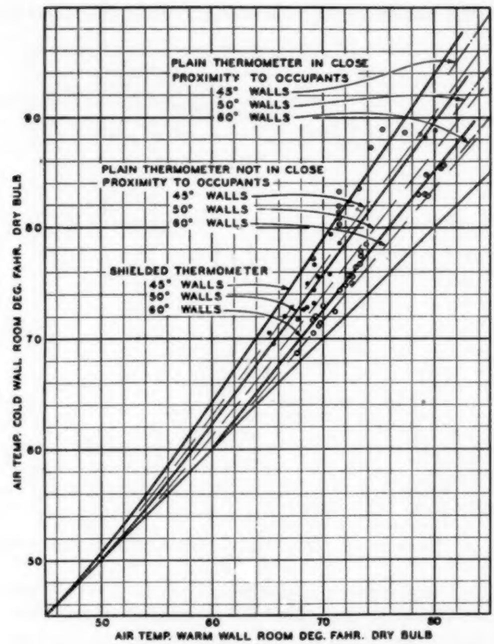


FIG. 3. RELATION BETWEEN DRY-BULB AIR TEMPERATURE IN WARM WALL ROOM AND DRY-BULB AIR TEMPERATURE IN COLD WALL ROOM FOR SAME FEELING OF WARMTH. TEMPERATURES AT 30 IN. ABOVE THE FLOOR INDICATED BY SHIELDED THERMOMETER, PLAIN THERMOMETER NOT IN CLOSE PROXIMITY TO PEOPLE, AND PLAIN THERMOMETER IN CLOSE PROXIMITY TO PEOPLE

earlier tests [3] made in determining lines of equal warmth for persons under different conditions of activity, clothing, and the cooling effect of air velocities. The subjects went alternately from the test room to the control room, remaining seated ten minutes in each, while the temperature in the control room and the relative humidity in both rooms were kept constant the temperature and moisture content of the air in the test room were varied between successive visits to it. At the start of a test the conditions in the test room were set to make the subject feel decidedly cooler than he felt in the control room. While the subject made successive alternate visits in the two rooms, the temperature

in the test room was gradually raised through the point where the subject expressed the same feeling of warmth in both rooms, and until he felt decidedly warmer in the test room. Thereupon the temperature in the test room was reduced in the same manner through the point of equal warmth and back to where the subject felt colder in the test room than in the control room.

During the test the subjects analyzed and recorded their own individual feelings of warmth or coolness independently of each other and without knowledge of the change in atmospheric conditions. Their judgment concerning their relative feelings of warmth in the two rooms was expressed as *colder, slightly colder, same, slightly warmer,* or *warmer,* referring in each case to the test room in comparison with the control room. In any single test the dry-bulb temperatures as indicated by the shielded thermometers, were recorded when each subject reported a feeling of *same.* These temperature readings were taken as the temperatures required to give equal feelings of warmth in the two rooms under prevailing conditions. In analyzing the results, the number of degrees higher, expressed in degrees dry-bulb, which it was found necessary to maintain the air in the test room over that in the control room in order to give the subject the same feeling of warmth, or the number of degrees higher in temperature it was necessary to raise the air in the test room to compensate for the effect of the cold walls, was taken as the cooling effect of the cold walls on the subject.

RESULTS

A number of curves giving the results of 51 tests according to individual reactions of the subjects are shown in Fig. 3, where the higher dry-bulb temperature in the test room is plotted against the dry-bulb temperature of the control room. In the studies made with test room walls at 45 F, 50 F, and 60 F, each respective cold wall temperature gives a band of points on the plot which fit the test data very well. The diagonal curve, representing equal temperatures in the test and control rooms, is also drawn. The number of degrees difference between the temperature indicated by this diagonal curve and the temperature of the test room, as given by the curve for each cold wall temperature, represents the cooling effect on a person for the particular wall temperature and the prevailing air temperature, or the number of degrees higher dry-bulb temperature which must be maintained in order to compensate for the excessive radiation loss to the cold walls of the test room.

The higher air temperature required to compensate for increased radiation loss in a cold wall room was expressed in deg dry-bulb instead of deg effective temperature for the reason that the effect is a function of the dry-bulb temperature of the air rather than of the effective temperature. The results of the study can be applied for any effective temperature by using the correction for the accompanying dry-bulb temperature in terms of effective temperature. Within the comfort zone, 1 deg change in the dry-bulb temperature of the air with no change in moisture content is equal to approximately 0.8 deg change in effective temperature.

The temperatures used as a basis for carrying on the study, as plotted in Fig. 3, are the shielded dry-bulb temperatures observed 30 in. above the floor

in the center of the rooms. Considerable variation in observed temperatures can be had in a room where there is a marked difference between the air and wall temperatures, depending upon whether the thermometer used has a plain or blackened bulb which is exposed directly to radiation, or whether the plain thermometer is shielded. Occupancy also affects temperature as read by the various types of instruments.

With an 80 F air temperature as indicated by the shielded thermometer two occupants in the room and with the wall temperature 45 F, the exposed thermometer with the plain bulb located at approximately the same point indicated a temperature of 78.4 F, and the exposed thermometer with a blackened bulb a temperature of 78.3 F. With other conditions the same but without the two occupants in the test room, the shielded thermometer indicated a temperature of 79.4, and the plain thermometer 77.4 F. It will be noted that the effect of the blackened bulb as compared with a plain bulb in indicating temperatures is negligible. The several instruments showed no measurable difference when located at the same point in the control room.

Shielding a thermometer reduces the effect of direct radiation between the instrument and the surrounding walls and objects. Such an instrument indicates very nearly the true dry-bulb temperature of the air at a particular point. Occupancy of the room has a two-fold effect on the temperature indicated by a thermometer: radiation from the surface of the clothed body raises the temperature of the bulb without raising the temperature of the air through which such radiation passes; and the occupants add sensible heat to the air, thus raising its true dry-bulb temperature. Hence, due to direct radiation from the subject, an unshielded thermometer when located in close proximity to a person registers a temperature higher than the true dry-bulb temperature of the air. A shielded thermometer, in an occupied room, however, indicates the true dry-bulb temperature of the air, so such temperature was accepted in this study as having the most practical significance.

Since room temperatures are usually measured with plain rather than shielded thermometers, it is of interest to consider the required temperature for comfort in a cold wall room as measured by a plain thermometer. Ordinarily, an unshielded thermometer is not placed as close to occupants as was necessary in the small test room, but rather far enough away to make the effect of radiation from the occupants negligible. In considering the desired temperatures in a cold wall room of normal size measured by an unshielded thermometer, it is more correct to consider the true temperature of the air as affected by the sensible heat from the occupants and radiation from the thermometer bulb to the cold walls, but not the effect of radiation from nearby occupants to the thermometer bulb. While such temperatures could not be read in the test room, they can be approximated by taking the temperature indicated by the plain thermometer in the unoccupied test room and correcting it for the true air temperature rise resulting from occupancy as indicated by the shielded thermometer. Curves based on this corrected temperature are shown in Fig. 3 as light-weight broken lines. The effect of the two occupants of the small test room on the temperature as indicated by the plain thermometer in close proximity to them is to give a higher temperature, though not as high a temperature as that indicated by the shielded thermometer. The relation between temperature in a cold wall room measured with a plain

thermometer located in close proximity to two occupants and that for the same thermometer corrected for close proximity is given at the high temperatures by the short broken line sections of the curves.

The data presented in the curves in Fig. 3 are again plotted for the purpose of practical application in Figs. 4 and 5, where the correction to be made on the temperature in a cold wall room in order that it may produce the same feeling of warmth as that of a warm wall room is given for various wall and air temperatures. The corrections shown in Fig. 4 apply when the temperature is observed by a shielded thermometer, and those shown in Fig. 5 apply when the temperature is observed by a plain thermometer as generally used in practice; more particularly, it will apply in such rooms when the plain thermometer bulb is not close to any person. If the thermometer is in close proximity to persons, as between seated persons in a class room, theater, or audience hall, then the correction due to radiation from the occupants as indicated in Fig. 3 should be taken into consideration.

Relation of the Data Presented to Actual Conditions in Ventilating Practice

The effect which radiation to cold walls will have on the desired temperature for comfort will vary with every installation, due to the variation in the ratio of cold walls to the total area of walls, floor and ceiling, and to the variation in the solid angle subtended by these cold surfaces at points on the occupants.

It is believed that the conditions under which the tests were made exaggerate the results, because few rooms have three cold walls, most having but one exposure. Hence, the results here presented should be considered as greater than usually met with in practice. On the other hand, if an extreme case should be met with, such as a sheet-metal cottage with all walls and roof exposed, the radiation effect would be much greater than indicated by test results; first, because of the greater effective area, and second because of the lower temperature which would pertain. Likewise, the effect of excessive window exposure may in special instances be greater. Window glass is ordinarily assumed to be opaque to low temperature radiation; therefore, the effect of a window should be the same as though it were a single thickness of sheet-metal. With zero F outside and assuming an outside film conductance of 5.6, an inside film conductance of 1.4, and 70 F air temperature indoors, the inside glass surface will be 12.9 F, which is much lower than the wall surface temperatures here studied. However, unless the glass area is excessively large, or unless the occupants are so close that the glass area subtends a large solid angle at points on their body surfaces, the effect will be comparatively small. Such conditions are so varied that it is obviously impossible in this paper to more than indicate their effect.

The close proximity of the subjects to the cold walls made necessary in the small test room may raise a question concerning the application of the findings to rooms of ordinary size. The effect which close proximity to the walls may have is two-fold; *first,* the solid angle subtended at the point of the occupant by the cold walls may be increased; *second,* downward convection currents of cold air near the cold walls may strike the subject.

The effect of close proximity of the subjects to the walls on heat loss by radiation depends only upon the change in the solid angle as discussed above, and since this depends upon both the distance from the wall and the size of the

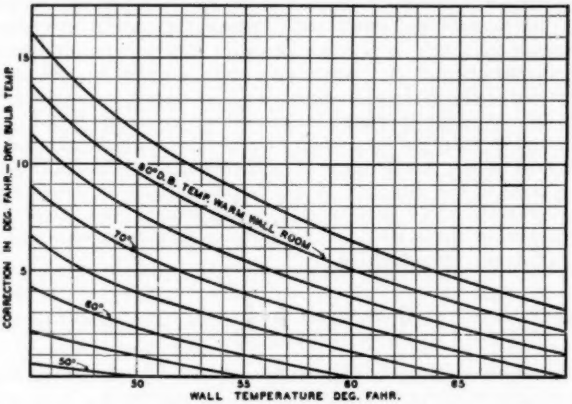


FIG. 4. CORRECTION TO VARIOUS DRY-BULB TEMPERATURES IN A WARM WALL ROOM FOR THE SAME FEELING OF WARMTH IN ROOMS HAVING THREE COLD WALLS. TEMPERATURES INDICATED BY SHIELDED THERMOMETERS 30 IN. ABOVE THE FLOOR

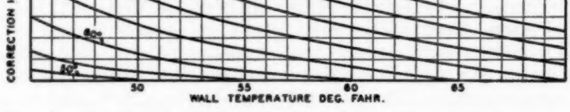


FIG. 5. CORRECTION TO VARIOUS DRY-BULB TEMPERATURES IN A WARM WALL ROOM FOR THE SAME FEELING OF WARMTH IN ROOMS HAVING THREE COLD WALLS. TEMPERATURES INDICATED BY PLAIN THERMOMETERS, 30 IN. ABOVE THE FLOOR, NOT IN CLOSE PROXIMITY TO OCCUPANTS

wall, this angle should not vary much as between the small test room and a room of ordinary size with the same proportion of cold walls in each.

A survey of temperatures in the horizontal plane at the 30-in. level showed a uniform temperature within plus or minus 0.5 deg, excepting for points distant 10 in. or less from the cold walls. Actually, a fall in temperature upon approaching the cold walls began at distances of approximately 15 in. At

10 in. from the wall this fall approximated 0.5 deg and increased as the wall surface was approached. The subjects were seated in the test room so that no part of their bodies was less than 12 in. from a cold wall surface. Hence, the cooling experienced in the small test room should not differ materially from like results for a larger room because of the closeness of the subjects to the walls.

During the study of the necessary compensating air temperature for cold walls, it was observed by the subjects that although they felt an overall sense of equal warmth in the test room to that experienced in the control room, local feelings of discomfort sometimes resulted for those parts of the body directly exposed to the cold walls having temperatures 50 F or lower. Discomfort might also result from a difference in temperature between the two sides of the subject's body and from his breathing air which is too warm. These experiences of local discomfort are common for a person seated near a large window in a warm room on a very cold day, unless a compensating radiant heat source is supplied from the same direction as the window. This is ordinarily taken care of in heating practice by placing a radiator under the window.

Since temperatures are observed ordinarily by plain mercury thermometers with bulb exposed to the same heat loss by radiation to the cold walls as are the surfaces of the clothed body, it might be assumed that the thermometer would automatically compensate for the greater heat loss by indicating an air temperature lower than that actually existing, so that when the thermometer indicates the desired temperature as given by the comfort zone, a true air temperature enough higher to compensate for the cold wall effect will result. The data obtained indicate that this is not entirely true, although the difference in temperature observed by the plain and shielded thermometers indicates some compensation in this direction. The reason for lack of complete compensation must lie in the difference in the surface area, the emissivity, the absolute temperature, or some combination of these existing between the thermometer bulb and the clothed body.

CONCLUSIONS

1. Increased radiation of heat from occupants in a room to cold walls on three sides is shown to require a higher air temperature for the same feeling of warmth. For example, an air temperature of 70 F with walls at the same temperature is equivalent to an air temperature 8.9 deg higher in a room having three of the four walls at 45 F. This effect is greater than usually occurs in practice for two reasons; *first,* few rooms have three exposed walls, and *second,* inside wall surface temperatures for reasonably good construction do not reach such low values for normal outside temperatures.

2. It is shown that in a cold wall room observed temperatures by mercury thermometers may vary depending on whether the bulb is shielded from or exposed to the *view* of the cold walls and of occupants.

3. Walls of large area having a considerably lower temperature than the surrounding air give a feeling of discomfort to occupants seated nearby due to the resulting feeling of coldness in those parts of the body exposed to the cold walls, even though the room air temperature is high enough to give an overall sense of not being cold. This is particularly true of large windows without a compensating radiator nearby.

DISCUSSION

A. C. WILLARD AND A. P. KRATZ (WRITTEN): This paper constitutes an important contribution adding to the more complete understanding of the effect of various factors on the requirements for human comfort.

In a paper[4] presented at the Summer Meeting of the Society, June 25, 1930, the writers emphasized the importance of considering wall surface temperatures in relation to comfort, but had no quantitative data indicating the amount of the correction required to be applied to the effective temperature. These data are now supplied by the paper of Messrs. Houghten and McDermott.

Reference to the paper on Wall Surface Temperatures indicates that with a standard frame wall in the room heating testing plant at the University of Illinois, an inside wall surface temperature of 55 F was obtained with 0 F outside and approximately 10 mph wind velocity. Hence, a temperature of 55 F cannot be regarded as unusual. Also, rooms with 3 wall exposures are not unusual. Under these conditions, Fig. 4 of the paper under discussion shows that at an indoor temperature of 70 F, a correction of 4 F is required to be added to the true dry bulb temperature in order to produce the same degree of comfort that would be obtained if both air and walls were at 70 F. This 4 F dry bulb is equivalent to 3.2 deg in the effective temperature.

Considerable attention has previously been directed toward the effect of relative humidity on human comfort. Reference to the comfort chart, however, shows that at 70 F dry bulb temperature, if the relative humidity is increased from a value of 20 per cent to one of 70 per cent, representing an increase of more than the width of the comfort zone, the effective temperature is increased only 3.5 deg. Comparing this with the 3.2 deg required with the cold walls, it is at once evident that a difference in temperature of 15 F between the air and the wall surfaces is equivalent to a change in relative humidity of 50 per cent, or more than the total range in relative humidity represented by the comfort zone. Hence, it would seem that, if anything, attention to the existence of and correction for cold walls is more important than attention to the maintenance of any stated relative humidity.

PROFESSOR WILLARD: I wish to call attention to the fact that in practically all heated rooms a considerable proportion of the outside wall surface consists of single sheets of window glass. The temperature of the inner face of these thin glass surfaces is very much lower than the temperature of the inner face of the wall surfaces, as reported by Messrs. Houghten and McDermott in their paper.

Even with absolutely no outside air movement the glass surface temperatures will be just about a mean between the inside and outside air temperatures. Hence, with an indoor-outdoor air temperature difference of 70 F, the inner surface of the window glass would be about 35 F instead of 55 deg, as used by Houghten and McDermott for their wall surface temperatures, but in reality, whenever there is any appreciable wind movement out-of-doors, the glass surface temperatures are very much less than the mean between the inside and outside air temperatures. For example, with a 10-mile wind velocity we have made, at the University of Illinois, actual glass surface temperature measurements in actual rooms with inside and outside air temperatures of 69.1 and —2.2 F, respectively, and have found that the actual temperature of the inner surface of the glass was 17.5 F. Hence, it will be apparent that a relatively small area of glass surface may be equivalent to a relatively large area of wall surface in affecting adversely the comfort of the occupants of the heated room during even moderately cold weather.

Moreover, these exceedingly cold glass surfaces are especially effective in increasing the air temperature differential between ceiling and floor and still further aggravating the conditions of personal discomfort.

[4] Wall Surface Temperatures, A. C. Willard and A. P. Kratz, A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 447.

No. 947

STUDY OF SUMMER COOLING IN THE RESEARCH RESIDENCE AT THE UNIVERSITY OF ILLINOIS

By A. P. KRATZ † AND S. KONZO ‡ (*MEMBERS*)
URBANA, ILL.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the *National Warm Air Heating Association*, and conducted at the University of Illinois, and is presented in collaboration with the paper entitled, Summer Cooling Operating Results in a Detroit Residence, by J. H. Walker and G. B. Helmrich

The results presented in this paper were obtained in connection with the summer cooling investigation 1932 at the Research Residence Fig. 1 at the University of Illinois, conducted by the Engineering Experiment Station of which Dean M. S. Ketchum is the director, under the supervision of A. C. Willard, Professor of Heating and Ventilation and head of the Department of Mechanical Engineering. These results will ultimately comprise part of a bulletin of the Engineering Experiment Station. Acknowledgment is due to M. K. Fahnestock, Special Research Assistant Professor, W. S. Harris, Special Research Assistant, E. L. Broderick, Research Assistant, and A. F. Hubbard, Research Graduate Assistant, for services rendered in connection with the investigation.*

THE principal objects of this investigation were (1) the determination of the cooling load and its hourly variation when cooling the Research Residence as a whole, under both night and day conditions; (2) the allocation to the various rooms of the heat entering the Residence, and the determination of the hourly variation in the cooling load of the individual rooms; and (3) the determination of the effectiveness of awnings as a means of reducing the cooling load of the Residence as a whole.

DESCRIPTION OF THE RESEARCH RESIDENCE AND COOLING PLANT

The Research Residence, shown equipped with awnings in Fig. 1, faces to the south and is of standard frame construction, with the exception that the

† Research Professor, Engineering Experiment Station, University of Illinois.
‡ Special Research Associate, Engineering Experiment Station, University of Illinois.
* The Research Residence in Urbana, Ill., was built, furnished and completely equipped specifically for research work in warm air heating, by the *National Warm Air Heating Association* in December, 1924.

Presented at the 39th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Cincinnati, Ohio, January, 1933, by S. Konzo.

studding is 2 in. x 6 in. instead of the 2 in. x 4 in. more commonly used, and that copper shingles are used on the roof. The wall section consists of weather boarding, sheathing building paper, studding, wood lath and plaster with rough sand finish. The coefficient of heat transmission for this wall section is 0.262 Btu per square foot per hour per degree Fahrenheit, at a wind velocity of 15 mph. The walls are not insulated, and no weather-stripping is used at the windows and doors. The total heated space in the winter, exclusive of the sun

FIG. 1. VIEW OF RESEARCH RESIDENCE IN URBANA, ILL.

room, is approximately 16,390 cu ft, and the heat loss at an outdoor-indoor temperature difference of 70 F is approximately 141,000 Btu per hour including basement loss.

For the summer cooling project, the third story was closed off by means of a door at the top of the stairs, and the sun-parlor by means of a door between it and the dining room. The total space cooled, therefore, consisted of three rooms on the first story, and three on the second story, together with the two interconnecting halls, making in all approximately 14,170 cu ft. The total calculated heat gain, exclusive of sun effect, for an outdoor-indoor temperature difference of 20 F, corresponding to the most severe day for the summer, was 33,500 Btu per hour. The kitchen was not used for cooking and no heat was present from this source.

All of the air used for cooling was recirculated, and the mechanical warm-air system used for heating the Residence in the winter, shown in plan in Fig. 2, was utilized as the distributing system to deliver the cooled air to the rooms, and to return the warmed air to the cooling coils and fan. The latter delivered 1,475 cu ft of air per minute. With one exception, the registers on the

delivery side were of the baseboard type. In the east bed room a wall type register was located 7 ft above the floor.

Part of the return air was by-passed through the cooling coils, as shown in Fig. 3, and was mixed with the balance of the return air on the suction side of the fan. The mixture was then delivered through the furnace casing into the distributing duct system. Control was obtained by means of a modulating by-pass damper operated from a room thermostat. Thus the temperature of

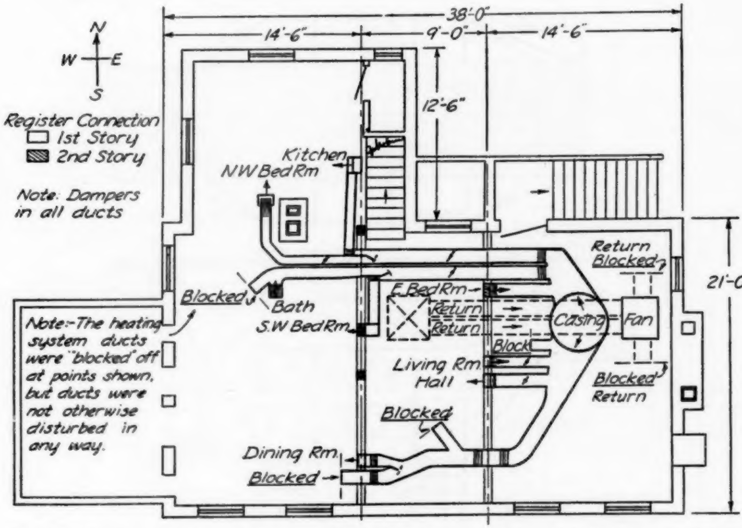


FIG. 2. BASEMENT PLAN SHOWING DUCT LAYOUT, FOR FORCED AIR SYSTEM, IN USE FOR DISTRIBUTING COOL AIR

the air at the registers was regulated to maintain the required dry-bulb temperature in the rooms.

For the purpose of these tests, a plant using ice as the cooling agent was selected in preference to one using mechanical refrigeration, because it was considered that the weight of ice melted could be conveniently and accurately determined, while the determination of the refrigeration load for the mechanical plant offered complications in testing that might be reflected in the accuracy of the results. No consideration was given to any possible relative merits in the two types of plants from commercial or other standpoints. The water circulated through the cooling coils was, therefore, cooled by means of ice.

Two sections of cooling coils, each section consisting of three rows of finned tubing set in headers, were used. The coils contained a total of 308 linear feet of ¾ in. tubing having helical fins, spaced 6 fins per inch. The overall diameter, including fins, was 1½ in. The inside dimensions of the casing were 27⅞ in. by 49⅛ in. Since the two coils were used in tandem, the

gross or *face* area presented to the flow of air was 9.5 sq ft, and the net free area was 4.2 sq ft. The total cooling surface was 473 sq ft. A drip pan at the bottom of the casing for the coils, as shown in Fig. 3, served to collect the condensation resulting from the dehumidification of the air. From the drip pan, the condensation was drained into a bucket placed on scales.

The arrangement of the cooling plant is shown in Fig. 4, and the details of the ice-tank in Fig. 5. The water was pumped from the bottom of the ice-tank

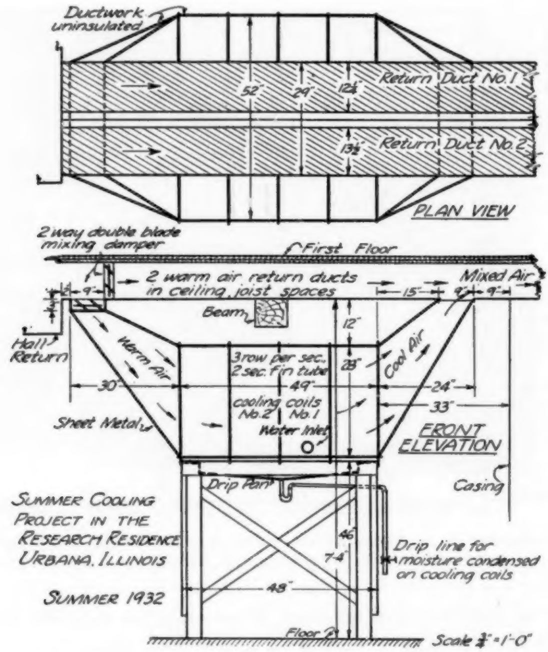


FIG. 3. DETAILS OF COOLING COILS IN MAIN UNIT

and passed through the cooling coils. The latter were connected in series, and the flow of the water was in the opposite direction to that of the air, thus forming a counter-flow arrangement. After passing through the coils, the water was returned to spray heads in the top of the ice-tank, and sprayed over the ice. The latter was placed on racks, and both the circulating water and the meltage accumulated in the bottom of the tank. This water space was connected to a weir box, shown as Section *B-B* in Fig. 5, containing a drain pipe, which faced upward with a sharp edge that served as a weir and maintained the water at a constant level just below the ice racks. All of the water that drained out of the weir box, therefore, represented the ice meltage that occurred during the time that the drain was open. By allowing this water to

collect in a pan on scales, it was possible to determine the ice meltage for any period of time desired.

For the purpose of these tests, the Research Residence was equipped with three temperature recorders which gave continuous records of the air temperature outdoors, at three points on the first story, at two points on the second story and at one point on the third story. In addition, a recording psychrometer gave a continuous record of the relative humidity on the first story.

For the work in the individual rooms, two commercial portable room coolers

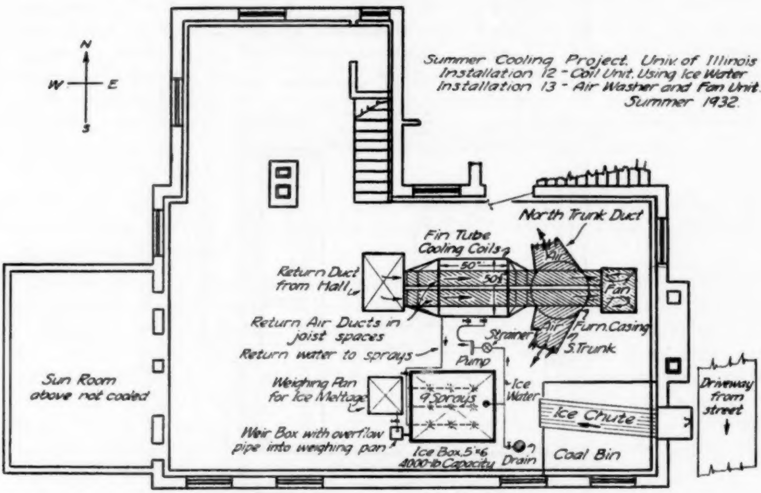


FIG. 4. LINE DIAGRAM OF ICE TANK, COOLING COIL UNIT, AND RETURN AIR DUCTS IN THE BASEMENT OF THE RESEARCH RESIDENCE, URBANA, ILL.

were used. These were placed on scales reading to 0.02 lb in order to determine the ice meltage and water resulting from dehumidification.

For part of the tests, the Research Residence was equipped with a total of 19 awnings, which were placed on the front, or south exposure, and on the two ends, or east and west exposures. The awnings were of the hood type with rope pull-up, consisting of a top and two wings with an 11 in. valance all around. They were made from 10 oz. duck, painted on one side only with gray and green stripes.

METHOD OF CONDUCTING TESTS

For a few of the preliminary tests, an attempt was made to maintain the schedule of desirable indoor temperatures corresponding to outdoor temperatures given as Table 2, page 10, in The A. S. H. V. E. GUIDE 1932. It at once became evident, however, that after the outdoor temperature had reached

a peak value and started to decline, it was necessary to progressively reduce the indoor temperature in order to maintain the schedule. This procedure imposed a load on the plant that was regarded as outside of the practical limits of operation. Furthermore, if the conditions indoors were conducive to comfort

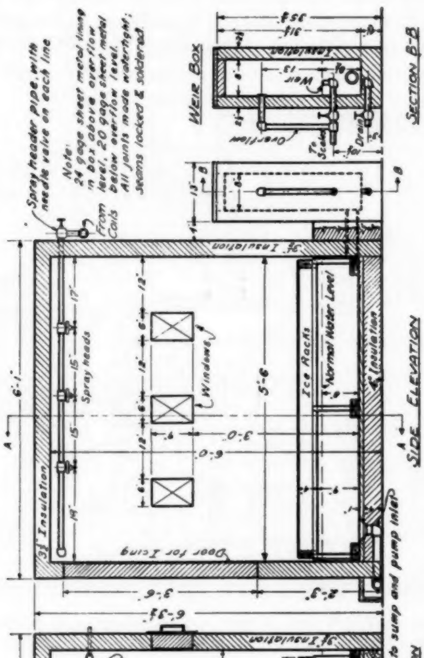


Fig. 5. Details of Ice Tank and Weir Box, Showing Arrangement of Ice Racks, Water Spray Pipes and Piping

during the peak outdoor temperature, there was nothing to indicate that the maintenance of these conditions resulted in discomfort after the peak had been passed.

The following method of operation was therefore adopted as a standard. The cooling plant was started when the effective temperature indoors rose to

75 deg, irrespective of the outdoor temperature. This occurred when the indoor dry-bulb temperature reached a value between 78 and 82 F, depending on the relative humidity. After starting the plant, the dry-bulb temperature was maintained approximately constant until the outdoor temperature dropped to about the same value. The plant was then stopped. Owing to the reduction in relative humidity after the plant had been in operation for about 1½ hours, the resulting effective temperature corresponding to the operating dry-bulb temperature of from 78 to 82 F was approximately 72 deg.

A calibration test on the plant, with the by-pass damper opened to pass the maximum amount of air through the cooling coils, indicated that if the temperature of the air at the registers was not allowed to fall below 60 F, 396 gal of water per hour had to be circulated through the coils. Since the plant was controlled with the by-pass damper and it was not possible, within the limits of the time available, to determine the optimum amount of water to be circulated under various conditions, all tests were run with 396 gallons of water circulated per hour at a pressure of 23 lb per square inch. The water entered the coils at about 35 F.

The dampers in the ducts were set to balance the plant with all of the room doors closed. When the plant was operated with the room doors open, the balance was not disturbed.

During the whole summer from June 1 to October 1 either continuous records or periodic readings were made of the temperature of the outdoor air, the temperature of the air in the basement and on each story, the temperature of the surface of the roof, the surface of the ceiling in one second story room, the relative humidity indoors and outdoors, and the times of starting and operating the plant. These readings were plotted on a continuous chart with time as a base, and to a scale of 0.4 in. per hour. This chart furnished a complete history of the variations in indoor and outdoor conditions, both during the periods of tests and the periods preceding the following tests.

For the heat load allocation tests, the unit room coolers were located in the rooms to be studied and these rooms were closed off from the rest of the Residence. The coolers were operated on the on-and-off principle to maintain the desired temperature in the rooms, and the central cooling plant was operated to maintain the same temperature in the rest of the rooms and halls. On account of the small amount of cooling weather obtained during the summer, the amount of data obtained with the unit coolers was entirely too meagre to establish any conclusion in regard to the allocation of the heat to the various rooms, or the hourly variation in the cooling load to these rooms.

General Results

In general, the results indicated that the ducts of a central forced-air heating system could be successfully adapted as a distributing system for cooled air without material alteration. Some readjustment of dampers was necessary in order to balance the plant when the cooled air was circulated. It was necessary to adjust the dampers so that more air was delivered to the second story than to the first. When the plant was balanced, the rooms were uniformly cooled with no indication of the existence of a pool of cold air near the floor, and

TABLE 1. COMPLETE SUMMARY OF TESTS

Date	Test No.	Start of Test	Length of Test Hours	Outdoor Temperature			In-door Temp. Aver.	Aver. Temp. Diff In—Out	Aver. Humidity		Ice Meltage		Dehumid-ification Total Lb	Hourly Heat Loads					
														Uncorrected		Corr. for Basement		Corr. for Basement	
				Max During Test	Min Dur-ing Day	Aver. Dur-ing Test			Out-door Aver. Dur-ing Test	Aver. In-doors	Equiv-Total Lb	Corr. for Base-ment Loss Lb		Total	Sensible	Total	Sensible	Total	Sensi-ble
														Btu per Hour				Btu per Degree per Hour	
7–7	P-4	10:45A	8.0	88.0	73.5	83.8	77.7	6.1	46.2	63.4	1060	701	32.19	19225	15000	12625	8400	2070	1377
7–9	P-5	1:15P	10.0	87.0	62.5	81.2	75.5	5.7	69.7	56.1	1459	1001	38.24	21000	16980	14400	10380	2526	1821
7–10	P-6	1:25P	4.0	90.5	70.5	86.3	77.2	9.1	65.6	60.4	472	289	20.51	16975	11590	10375	4990	1140	548
7–11	P-7	2:00P	6.0	85.0	66.5	81.5	78.2	3.3	47.4	53.3	907	632	27.82	22885	18010	16285	11410	4937	3456
7–12	P-8	2:00P	7.5	93.5	59.5	86.4	77.0	9.4	58.1	51.9	1318	975	41.40	25300	19500	18700	12900	1989	1372
7–13	P-9	9:30A	13.5	99.0	70.0	91.0	78.4	12.6	58.0	50.8	2642	2024	97.96	28180	20560	21580	13960	1712	1108
7–14	P-10	11:40A	14.25	99.0	75.0	89.4	81.1	8.3	62.8	49.2	3050	2397	113.23	30820	22470	24220	15870	2917	1912
7–15	P-11	10:00A	24.0	103.0	78.0	88.8	80.8	8.0	59.9	46.2	4243	3144	111.56	25450	20400	18850	13800	2356	1725
7–16	P-12	10:00A	24.0	97.0	77.5	83.2	80.0	3.2	59.1	45.1	2713	1614	76.48	16280	12930	9680	6330	3026	1978
7–17	P-13	10:00A	12.0	90.5	72.5	85.8	80.7	5.1	37.3	44.0	1261	711	30.31	15130	12480	8530	5880	1673	1153
7–20	P-14	9:00P	24.0	98.0	78.0	86.7	80.0	6.7	60.5	42.9	3530	2431	109.04	21170	16410	14570	9810	2174	1464
7–21	P-15	9:00P	24.0	97.5	74.0	83.9	79.2	4.7	55.4	43.3	3320	2221	88.94	19920	16030	13320	9430	2833	2005
8–10	P-20	9:30A	12.0	91.0	67.0	83.1	76.4	6.7	73.4	48.8	1761	1211	61.11	21120	15775	14520	9175	2166	1369
8–15	P-21	9:30A	10.5	87.0	62.0	81.8	75.6	6.2	71.5	50.4	1238	757	35.49	16975	13425	10375	6825	1674	1101
8–16	P-22	9:00A	13.5	90.0	72.0	84.0	77.1	6.9	76.3	48.6	2025	1407	60.71	21600	16880	15000	10280	2173	1489
8–23	P-24	10:00A	11.0	87.5	57.5	80.9	74.9	6.0	52.5	44.4	1838	1334	40.64	24060	20180	17460	13580	2908	2262
8–24	P-25	10:00A	11.0	90.0	60.0	83.5	75.1	8.4	52.3	46.0	1464	960	35.01	19170	15830	12570	9230	1496	1098
8–25	P-26	9:00A	13.0	89.5	69.0	83.6	77.3	6.3	63.5	48.1	1578	983	45.01	17470	13835	10870	7235	1725	1148
8–26	P-27	9:00A	12.0	87.0	67.5	80.8	76.6	4.2	69.0	46.5	1513	963	41.44	18155	14525	11555	7925	2751	1886
8–29	P-28	7:00A	24.0	94.0	75.0	82.7	78.0	4.7	74.2	47.2	3680	2581	104.77	22090	17500	15490	10900	3295	2318
8–30	P-29	7:00A	24.0	95.0	73.0	81.9	77.9	4.0	72.2	47.4	3752	2653	109.39	22500	17720	15900	11120	3975	2780
8–31	P-30	7:00A	24.0	93.0	71.5	79.0	77.1	1.9	74.1	45.9	3237	2138	87.15	19420	15600	12820	9000	6750	4737

as the sun changed position, the balance was not materially disturbed when the plant was controlled with a single thermostat placed in the dining room. An average temperature difference of approximately 3.5 deg was observed between the floor and the ceiling, and 1.8 deg between the floor and the breathing level, with the cooler air near the floor. The air was delivered from the registers at temperatures varying from 60 to 70 F, depending on the outdoor temperature, and at average velocities varying from 50 to 450 fpm measured with an anemometer placed one inch from the register face. No objectionable drafts were observed in the rooms, except in one case in which the air was delivered from a baseboard register at a velocity of approximately 450 fpm. Even in this case, conditions were easily corrected by employing a baffle in front of the register face in order to direct the air flow toward the ceiling. The best distribution of cooled air in the room was obtained with the wall register placed 7 ft above the floor. In this case, the velocity of the air leaving the register was approximately 350 fpm. The gain in temperature of the air, from the fan inlet to the register faces did not exceed 3 deg, and condensation did not appear at any time on the furnace casing or basement duct system. No insulation was used on these surfaces. The basement temperature remained practically constant at 70 F.

The cooling season at Urbana, Illinois, during the summer of 1932, extended from June 1 to October 1, with a total of 62 days on which the maximum temperature reached 85 F or more. This represented a total of 1,471 degree-hours above 85 F. A total of 43.3 tons of ice was used for the season. The maximum hourly rate of ice meltage observed was 220 lb. This rate occurred on Test P-29, when the Residence was not equipped with awnings, and for which the maximum outdoor temperature attained was 95 F. The plant had been in operation for the previous 24 hours. The maximum meltage for any 24-hour period was 4,243 lb. This occurred on test P-11, when the Residence was equipped with awnings, and for which the maximum outdoor temperature attained was 103 F. In this case, however, the plant had not been in operation for the 24 hours previous to the test.

Since the total volumes of air and water circulated were not varied for the different tests, the electrical input to the fan and pump motors remained practically constant. The power consumed by the fan motor was ¼ kw, and by the pump motor was 1¼ kw. The power required by the pump was influenced by the pressure of 23 lb per square inch required to operate the spray heads in the ice tank. The use of these spray heads was found necessary in order to break the water up into a sufficiently fine spray to prevent the formation of craters in the ice. These craters filled with water and interfered with the accuracy in the determination of the ice meltage. The gain in accuracy resulting from the elimination of the craters, however, was obtained by a sacrifice in the operating efficiency of the plant as a whole, due to the pressure required by the spray heads. In a service plant, where the accurate determination of the ice meltage is not necessary, the spray heads could be dispensed with and the water showered over the ice in larger streams, thus reducing the power for the pump.

The cooling load in the basement consisted of the heat gain of the coils, furnace casing and duct system, and the electrical load imposed by the fan and pump motors and the lights. Separate determinations of this load indicated

that it remained practically constant at 45.8 lb of ice per hour, or 6,600 Btu per hour.

The principal results for all tests are given in Table 1. It has not been possible to use all of these for the curves and analyses. The lengths of tests

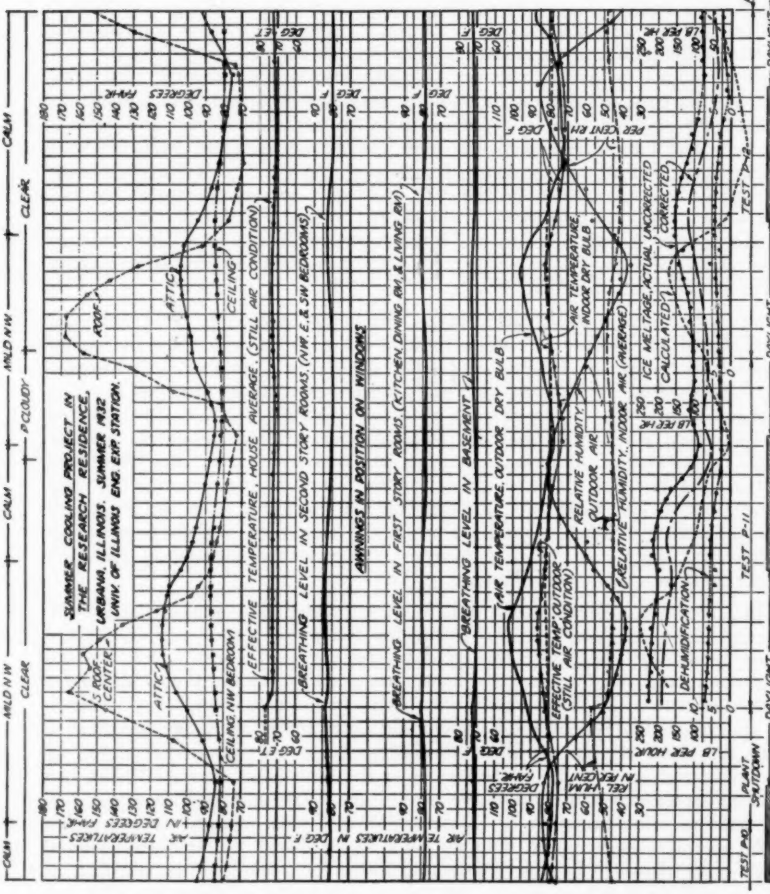


Fig. 6. Graphic Log of Observations—July 15, 16 and 17, 1932

were necessarily determined by the character of the weather. Furthermore, the total ice meltage in the case of short tests was largely influenced by the history of the Residence previous to the start of the test. On some of the shorter tests, the starting load formed an excessive proportion of the total ice meltage for the test. Hence, many inconsistencies may be observed in compari-

sons based on the averages and totals for individual tests. It was therefore necessary, for the purpose of analysis, to group the tests made under comparable conditions, and in some cases the analysis has been based on the maximum load extending over a 4-hour period including the maximum outdoor temperature. Some comparisons have been based on the load corrected for basement losses, since this represented the part of the load influenced by the particular factor under consideration. Furthermore, this basement load would vary for different plants and, by especial attention to features of design favoring higher operation efficiency rather than adaptability to accurate measurement of test data, could be very materially reduced.

In the test plant, the basement load was augmented by the necessity for the use of spray heads in the ice tank and by the use of more basement lights than would be demanded by a service plant. Also, during the tests, an effective temperature of approximately 72 deg was maintained in the Residence. This condition was perfectly comfortable with the occupants clothed in light summer clothing. No member of the staff experienced any ill effect either from shock due to sudden changes in environment or from contracting a cold. However, the indications were that somewhat less cooling would be satisfactory for residence service. Just how much less is debatable at present. Hence, it is possible that by varying the design and operation from that used for the test plant, a reduction in ice meltage could be made. The figures presented, however, may be considered as representative, with a resonable margin of safety, of the cooling requirements for a structure similar to the Research Residence.

A complete graphic log covering a period of two 24-hour tests with the house equipped with awnings is shown in Fig. 6, and a similar graphic log for two 24-hour tests with the awnings removed is shown in Fig. 7. Fig. 6 contains all of the data from the corresponding portion of the continuous chart previously mentioned, with the addition of effective temperatures, ice meltage and calculated cooling load. In Fig. 7, the data on roof, attic, and ceiling temperatures have been omitted.

From Fig. 6 it is interesting to note that the temperature of the copper shingles on the roof attained a value as high as 169 F. At times values as high as 175 F were observed. The temperature of the air in the attic space rose to 112 F. The time shown between the attainment of maximum temperature for the roof surface and that of maximum air temperature in the attic does not represent the time lag of the attic space, however. The data for the roof were taken from the south slope, while the attic space was in a north wing, and the west slope of the roof above it did not receive the sun until afternoon. The time at which the maximum was attained in the attic space corresponded approximately with that for the maximum outdoor temperature, thus indicating comparatively no lag. The temperature of the surface of the ceiling below the attic reached a maximum about 3 hours later than that of the attic itself. In this case, the ceiling was insulated with one inch of insulating quilt, and the surface temperature never rose more than approximately 3 deg above the temperature of the air in the room below it. The fact that the ceiling below an attic space becomes in effect a large panel radiator, thus adding to the discomfort of the occupants of the room independently of the air temperature, indicates a necessity for effective insulation outside of the desirable reduction in cooling load accompanying its use.

RELATIVE HUMIDITIES AND DEHUMIDIFICATION

Fig. 6 illustrates the effect of the indoor relative humidity on the starting load on the plant. The plant was not operated on the day previous to the start

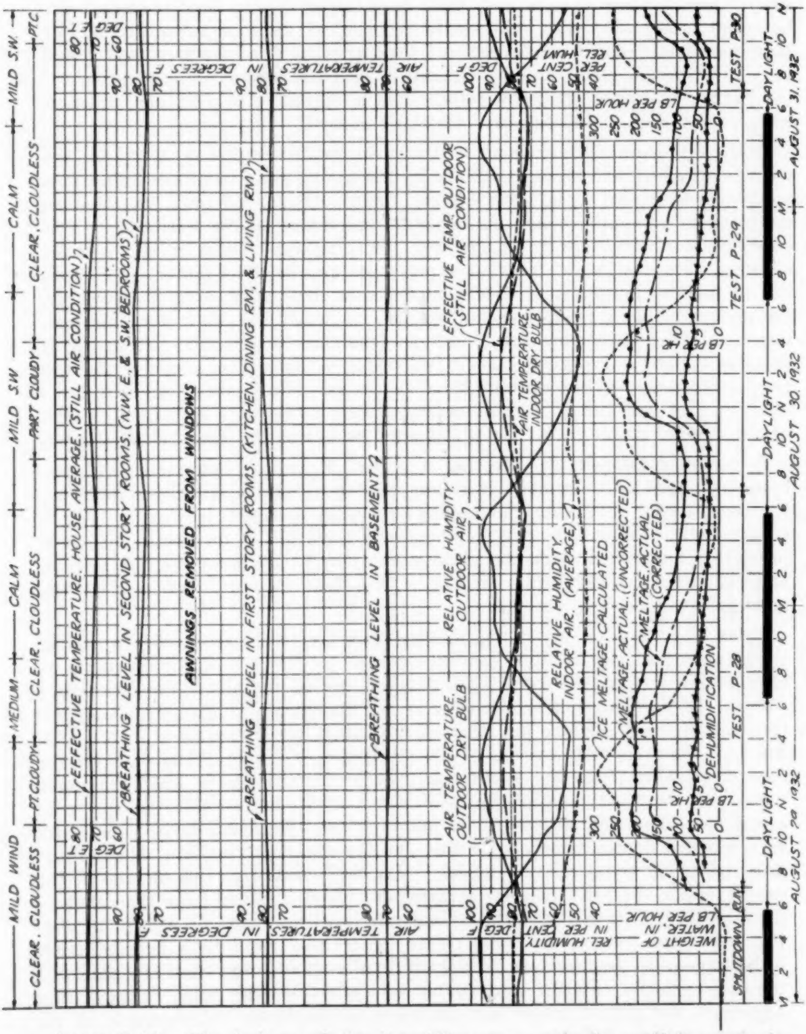


FIG. 7. GRAPHIC LOG OF OBSERVATIONS—AUGUST 29, 30 AND 31, 1932

of test P-11. The indoor relative humidity rose gradually until it reached a value of 57 per cent at 10 a. m., July 15, at which time the outdoor temperature

was 92 F and the indoor temperature was 81 F, corresponding to an indoor effective temperature of 76 deg. At this time the cooling plant was started for test P-11. At the end of an hour's operation, the relative humidity had dropped to about 46.5 per cent, and at the end of four hours had stabilized at about 45 per cent. Meanwhile the indoor effective temperature dropped from 76 deg to 72 deg while the indoor dry-bulb temperature remained comparatively constant at about 81 F. The dehumidification, appearing as condensation on the coils, decreased from about 7 lb per hour to 5.5 lb per hour. Data on other tests showed decreases from as high as 9 lb per hour to as low as 4.5 lb per hour within two hours after the start of the test. The average dehumidification load ranged from approximately 3,000 to 8,000 Btu per hour, and formed a considerable part of the total load. It averaged about 4,000 Btu per hour, and constituted approximately one-fifth of the total load. A number of calculations, based on the condensation weighed after the indoor relative humidity had stabilized, and at outdoor temperatures near the peak, gave results for the number of air changes ranging from 0.7 to 1.0 air changes per hour. These results were subject to changes in moisture content of the walls, floors, and furniture, but the consistency of the weighed amounts, and the fact that long periods of uniform indoor relative humidities were maintained, indicates that three-quarters of an air change per hour represents the amount of air infiltration with a fair degree of accuracy.

COMPARISON OF ACTUAL AND CALCULATED COOLING LOADS

The variation in the actual cooling load, as determined by ice meltage from hour to hour, may be observed from Figs. 6 and 7. From these curves it may be noted that while the outdoor air temperature reached a maximum at about 4 p. m., the actual load reached a maximum from two to four hours later. This was particularly true if the plant had been operated during the 24 hours previous to the start of a test. The effect of the sun on the load may be observed by comparing the shape of the load curves in Figs. 6 and 7. In the case of Test P-12, shown in Fig. 6, for which the Residence was equipped with awnings, the load increased gradually to a peak occurring four hours later than the time for the maximum temperature for the day. In the case of Tests P-28 and P-29, shown in Fig. 7, the load rose suddenly, reaching one peak at about noon, then declined somewhat and increased to reach a second peak at 5:30 p. m. The first peak was probably caused by the sun, which showed its maximum effect at 10 a. m., and the second peak by a combination of the sun, which reached a secondary maximum at 2 p. m., and the heat lag of the structure. The total heat radiated through the windows, based on the glass area and exposure of the windows and on the curves given by F. C. Houghten,[1] and others was as follows: 6 a. m., 980; 8 a.m., 11,587; 10 a. m., 15,040; 12 n., 11,290; 2 p. m., 14,065; 4 p. m., 10,093; 6 p. m., 828 Btu per hour. A limited number of observations on the solar radiation, a sample of which is shown in Fig. 8, indicated that the solar radiation at Urbana, Illinois, which is in practically the same latitude as Pittsburgh, Pennsylvania, agreed reasonably well with the results obtained at Pittsburgh by the authors previously cited.[1] Their curves were therefore accepted as applying at Urbana. The observations

[1] Heat Transmission as Influenced by Heat Capacity and Solar Radiation, by F. C. Houghten, J. L. Blackshaw, E. M. Pugh, and Paul McDermott, A. S. H. V. E. TRANSACTIONS, Vol. 38, 1932, p. 231.

of solar radiation were made with an Abbot silver disk pyrheliometer,** a description of which is given by J. H. Walker [2] and others.

A comparison of the actual cooling load, based on measured ice meltage, and the calculated cooling load reduced to terms of ice meltage, may be made from Figs. 6 and 7. This comparison should be based on the cooling load corrected for loss in the basement, since the calculated load applies only to the portion of the Residence above the basement. The calculated load at any given time consisted of the sum of the sun load, the dehumidification load, the load due to occupancy, the load due to sensible heat from air infiltration, and the heat transmission load. The latter was based on the actual outdoor-indoor temperature difference shown by the curves at that time and the heat transmission coefficients applying to the walls, windows, floors and ceilings of the Residence. The sun load was based on the glass exposures for the Residence and on the solar radiation obtained from the curves of Houghten [1] and others corresponding to the given time, with the assumption that a single thickness of glass intercepted 17 per cent of the total radiation. The latter was determined experimentally and checked closely with the figure given by Walker [2] and others. In the case of the Residence equipped with awnings, the latter were considered as completely intercepting the sun's radiation, thus eliminating the sun load caused by radiation through the windows. The dehumidification load and the load due to sensible heat in air infiltration were based on the existing indoor and outdoor air temperatures and relative humidities, and on the assumption of three-quarters of an air change per hour for the 14,170 cu ft of space cooled. Occupancy was assumed to be between two and three people, or 1,600 Btu per hour.

From these curves, it is evident that the heat lag of the structure operates in such a way as to make it almost futile to attempt to calculate the cooling load for the building at any specified time. The actual load never attained a value as high as the calculated maximum load corresponding to the maximum outdoor temperature. Furthermore, the true load at the time that the maximum load actually occurred, was much greater than the calculated load corresponding to this time. The curves all showed one point in common, however. That is, the calculated load at approximately 6 p. m. agreed with the actual load at this time, and corresponded to a value only slightly less than the actual maximum load. It is therefore possible that the maximum load for the purpose of design, could be based on the sun effect and the probable outdoor temperature at 6 p. m. for any given locality. Whether or not this is a feasible method of calculation, and whether it would apply to structures differing in character from the Research Residence is admittedly somewhat problematical.

The fact that the area between the curves in excess of the actual load is approximately equal to the area representing the deficiency over actual load, suggests the possibility of obtaining an average load by calculation, based on the average temperature differences, relative humidities and sun effect spread over 24 hours. This average sun effect, consisting of the total Btu during the hours of sunshine divided by 24, was 5,300 Btu per hour, or 35 per cent of the maximum. For Test P-29, the calculated average cooling load, based on 24 hours, was equivalent to 116 lb per hour of ice meltage. The actual average

** Loaned by the Smithsonian Institution, Washington, D. C.

[2] Field Studies of Office Building Cooling, by J. H. Walker, S. S. Sanford and E. P. Wells, A. S. H. V. E. TRANSACTIONS, Vol. 38, 1932, p. 285.

ice meltage was 111 lb per hour. Hence, the calculated meltage was 4.5 per cent in excess of the actual. This method has no merit as a means of determining the maximum load, since the maximum hourly meltage for this test was 220 lb per hour. However, if the seeming incongruity of considering the sun load as distributed uniformly over 24 hours is disregarded, the method might have a possible application in estimating the probable daily meltage.

Effect of Outdoor Temperature on Cooling Load

An attempt to correlate the cooling load with the average outdoor-indoor temperature differences shown in Table 1 resulted in many inconsistencies, due

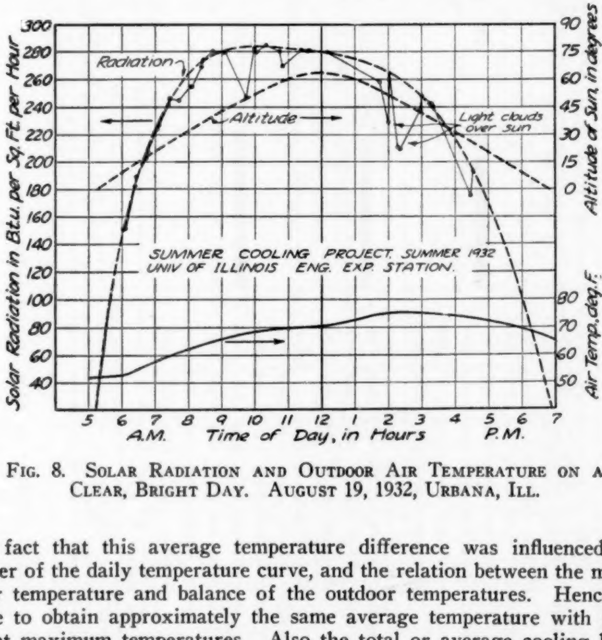
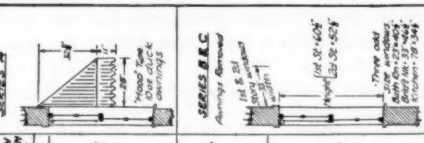


Fig. 8. Solar Radiation and Outdoor Air Temperature on a Clear, Bright Day. August 19, 1932, Urbana, Ill.

to the fact that this average temperature difference was influenced by the character of the daily temperature curve, and the relation between the maximum outdoor temperature and balance of the outdoor temperatures. Hence it was possible to obtain approximately the same average temperature with radically different maximum temperatures. Also the total or average cooling load was dependent on the heat lag and previous history of the structure, and on the portion of the day over which it was possible to run the test. Therefore, a number of tests were selected in which the start of the test occurred a sufficient length of time previous to that for attainment of maximum outdoor temperature to permit conditions to become stabilized before the maximum outdoor temperature was reached. For this group of tests, the cooling load per degree difference in temperature was based on the average outdoor-indoor temperature difference for the four hours included between two hours before and two hours after the time for the maximum, or peak temperature. These results are tabulated in Fig. 9 and have been plotted against the average temperature over the four hour peak period in Fig. 10.

From Fig. 10 it is evident that the cooling load is not directly proportional to the difference in temperature, but for the same difference in temperature the cooling load increases as the outdoor temperature increases. This was true

TEST NO.	DATE 1932	LENGTH OF TEST	WEATHER: SKY	WEATHER: SUN	WEATHER: WIND	AVER. TEMP. OVER 4 HOUR PEAK PERIOD: OUTDOOR AIR	AVER. TEMP. OVER 4 HOUR PEAK PERIOD: INDOOR AIR	B.T.U. PER HR. PER DEG. DIFF. IN OUTDOOR-INDOOR TEMP.: UNCORRECTED FOR BASEMENT: TOTAL	UNCORRECTED FOR BASEMENT: SENSIBLE	CORRECTED FOR BASEMENT: TOTAL	CORRECTED FOR BASEMENT: SENSIBLE	AVER.-CORRECTED FOR BASEMENT (0 400 800 1200 1600)	RATIOS OF COOLING LOAD: S	RATIOS OF COOLING LOAD: T	PERCENT REDUC. IN LOAD WITH AWNINGS: S	PERCENT REDUC. IN LOAD WITH AWNINGS: T
SERIES A. AWNINGS IN POSITION; CLEAR BRIGHT DAY																
9	7-13	13.5	Clear; few Cum.	Bright	Mild	97.5	78.8	1650	1192	1297	840					
11	7-15	24.0	Clear	"	"	101.3	81.9	1660	1330	1320	990	T 1234		0.79		21
12	7-16	24.0	"	"	"	94.8	81.8	1627	1282	1120	775	S 853	0.78		22	
14	7-20	24.0	Few Cum.	"	"	95.5	80.7	1646	1254	1199	808					
				Average				1646	1265	1234	853					
SERIES B. AWNINGS REMOVED; HAZY DAY WITH SCATTERED GREY NIMBUS CLOUDS																
21	8-15	10.5	Cldy	Dull	Calm	84.9	76.7	2273	1779	1468	975	T 1458		0.93		7
26	8-25	13.0	"	"	Med.	87.5	77.9	2135	1661	1448	974	S 975	0.90		10	
				Average				2204	1720	1458	975					
SERIES C. AWNINGS REMOVED; BRIGHT DAY WITH SCATTERED WHITE CUMULUS CLOUDS																
22	8-16	13.5	Part Cldy with few white cumulus clouds	Bright but sun in and out	Calm	89.3	78.0	2107	1638	1524	1055					
27	8-26	12.0			Med.	85.8	76.9	2205	1742	1464	1000					
28	8-29	24.0			Mild	93.1	79.0	2063	1608	1596	1140	T 1561		1.00		0
29	8-30	24.0			Med.	94.2	79.8	2167	1612	1708	1153	S 1087	1.00		0	
30	8-31	24.0			Med.	91.9	78.5	2004	1582	1512	1088					
				Average				2109	1636	1561	1087					



NOTES: Awnings on all East, South, and West windows, 17 in number. Windows closed and locked; window shades drawn half way down.
Total cooling load is the sum of the "dehumidification load" and the "sensible heat load."
Basement correction for heat load includes heat losses in basement from ice tank, coils, piping, etc.
Values in above table were based on results obtained during the 4-hr. period of max. outdoor temperature.
All 4-hr. peak periods were preceded by several hours of actual plant operation.

FIG. 9. EFFECT OF AWNINGS ON REDUCTION OF COOLING LOAD

both for the sensible and the total cooling load, the latter including dehumidification, and whether or not the house was equipped with awnings. This is equivalent to a statement that the cooling load per degree temperature difference is not constant, and that any method of calculation based on Btu per degree-

hour and seasonal degree-hours is incorrect unless some account is taken of the increase in cooling load per degree difference in temperature as the actual outdoor temperature increases.

EFFECT OF AWNINGS

Fig. 9 shows the reduction in cooling load effected by the use of awnings on all east, south, and west exposures. This comparison has been made between the cooling loads corrected for basement loss, since the only portion of the load affected by the awnings was that above the basement, and the base-

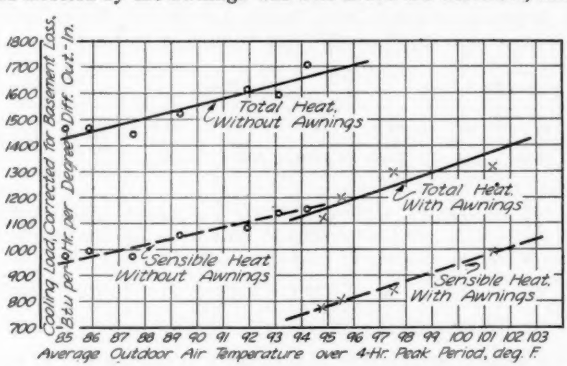


FIG. 10. VARIATION IN COOLING LOAD WITH OUTDOOR AIR TEMPERATURE DURING 4-HOUR PEAK PERIOD

ment load would vary with different individual plants. Comparing the averages for the two groups of tests run on bright, practically cloudless days, it may be noted that the awnings effected a reduction in sensible cooling load of 22 per cent and in total cooling load of 21 per cent. It may be noted that the tests without awnings, P-21 to P-30 inclusive, were run nearly one month later than the ones with awnings. A calculation of the total amount of heat from the sun radiated through the windows, based on the curves of Houghten [1] and others, indicates that the heat from this source is somewhat greater during September than during August. Since the awnings intercept all of the sun load imposed through the windows, it is immaterial whether these tests were run during August or during September. However, the load obtained without awnings during September was probably somewhat greater than it would have been if the tests had been run during August. The comparison that has been made, is equivalent to assuming that both series of tests were run during September. If it had been possible to compare two series of tests made in August, the load with awnings would have been the same, but the load without awnings would have been somewhat less than the one actually used. To this extent therefore the comparison as made has tended to favor the awnings. To offset this, however, the average outdoor temperature was about 6.5 deg higher during August, for the group of tests with awnings than during September for the one without awnings, and since the cooling load per degree difference

in temperature increases with the outdoor temperature, the comparison as made has tended to be less favorable to awnings. This is illustrated in Fig. 10, from which it may be noted that the reduction in cooling load resulting from the use of awnings is approximately 30 per cent for outdoor temperatures between 93 and 97 F. Hence, it is evident that the actual percentage of saving effected by awnings is dependent on what part of the season is selected for comparison, but it is probable that the figures based on the comparison as made are conservative for the average reduction of load resulting from their use. It is possible that the results may be affected by the variations in the sun

TABLE 2. SUMMARY OF TOTAL DAYS, HOURS AND DEGREE-HOURS ABOVE BASE TEMPERATURES OF 85, 90 AND 95 F FOR SUMMER OF 1932 at URBANA, ILL.

Month	Total Days			Total Hours			Total Degree-hours		
	85 F and Above	90 F and Above	95 F and Above	85 F and Above	90 F and Above	95 F and Above	Above 85 F	Above 90 F	Above 95 F
June.....	18	5	...	72.0	16.6	...	222.8	30.5	...
July.....	22	14	8	163.8	81.7	29.7	929.9	327.8	72.9
August...	18	6	...	90.6	23.5	...	316.3	50.8	...
Sept.....	4	...	...	2.7	...	...	1.7	...	...
Total..	62	25	8	329.1	121.8	29.7	1470.7	409.1	72.9

effect on the walls from month to month, but it is not possible to separate this for the purpose of analysis.

Referring to Figs. 6 and 7, it may also be noted that the awnings changed the character of the daily cooling load curve, and reduced the extent of the period of maximum cooling load from a duration of approximately 6 hours to one of 2 hours.

SEASONAL COOLING LOAD

Table 2 gives a summary of the number of days, hours, and degree-hours above bases of 85, 90 and 95 F for the summer of 1932 at Urbana, Illinois, based on data observed at the Research Residence. From this it is evident that the choice of a base temperature is an important item in determining the number of degree-hours in a cooling season. If it is assumed that no cooling will be required until the outdoor temperature reaches 90 F then the summer of 1932 would have had a total of 409.1 degree-hours requiring cooling. If, however, a base of 85 F is assumed, then the summer would have had a total of 1470.7 degree-hours, or 3.6 times the amount for a 90 F base.

The uniformity of the curves of outdoor temperature plotted against the time of day as in Figs. 6 and 7 suggested the possibility that there might be some correlation between the maximum daily temperatures and the number of hours and degree-hours above any given base temperature. The daily outdoor temperatures, in general, followed a cycle similar to the one shown as an inset in Fig. 11. Therefore, if a given base temperature was selected, the length of the cord between the two points at which this base temperature line intersected the curve of outdoor temperatures, represented the number of hours for

the given day that temperatures at, or above, the base temperature persisted. Also, the area between the base line and the temperature curve above the base line, shown cross-hatched in Fig. 11, represented the number of degree-hours above the given base. This procedure was followed for each of the 62 days during the summer of 1932 for which the maximum temperature reached 85 F or above, using three separate base lines, at 85, 90 and 95 F. The resulting number of hours and degree-hours for each day were then plotted against the maximum outdoor temperature for that day, as shown in the two groups of curves in Fig. 11. It was then possible to draw three curves, for base temperatures of 85, 90 and 95 F, representing the average of the points with a fair degree of accuracy. The points accordingly represent the deviations of the actual cycles for the individual days from the ideal average cycle represented by the curves. As an example, if a base temperature of 85 F is selected, and the maximum temperature of 92 F was attained on a given day, then from the upper group of curves in Fig. 11, a total of 7.2 hours with temperatures above 85 F could be expected for that day, and from the lower group of curves, a total of 30 degree-hours would be probable. There is no reason to believe that the character of the daily temperature cycles used in constructing these curves is peculiar to Urbana, Illinois, alone, and therefore it is probable that the curves in Fig. 11 are applicable to a wide range of localities. Hence, it should be possible to estimate the probable number of hours and degree-hours above a selected base for any season in a given locality, by counting the total number of days occurring at various maximum temperatures from the U. S. Weather Bureau reports for the locality, and multiplying the number of days thus obtained for each maximum temperature by the number of hours or degree-hours obtained from the curves in Fig. 11 corresponding to the various maximum temperatures selected. The summation of the hours or degree-hours thus obtained represents the total for the season.

An estimation of the number of degree-hours above 85 F for each of the past ten summers at Urbana, Illinois, has been made in this manner, and is shown in Table 3. In compiling this table, the various ranges of outdoor temperatures have been grouped as shown in Column 1. The mean temperatures for the various ranges are given in Column 2, and the number of degree-hours per day, as read from the curves in Fig. 11 corresponding to the mean temperatures in Column 2, are given in Column 3. In the two columns for each year shown, the first column is the total number of days included in the months of June, July, August, and September, having maximum temperatures falling between the limits shown in Column 1, and the second column is the product of the number of days and the corresponding number of degree-hours per day from Column 3. The first three columns in this table may be considered as applicable to any locality, while the balance of the table applies to Urbana only. The ranges could be made smaller than those shown in Column 1, but greater accuracy at this point is hardly warranted when it is considered that the final result is only a reasonably close approximation at best.

That it is a reasonably close approximation is indicated by the fact that the actual number of degree-hours for the summer of 1932, calculated from the summation of the degree-hours for the individual days was 1,471, while the number calculated from the average curves and shown in Table 3 was 1,538, or a difference of 4.2 per cent. A similar approximation for the number of hours

above 85 F gave 318 as compared with the actual number of 329, or a difference of 3 per cent.

From Table 3 it is evident that the cooling season is extremely variable, even

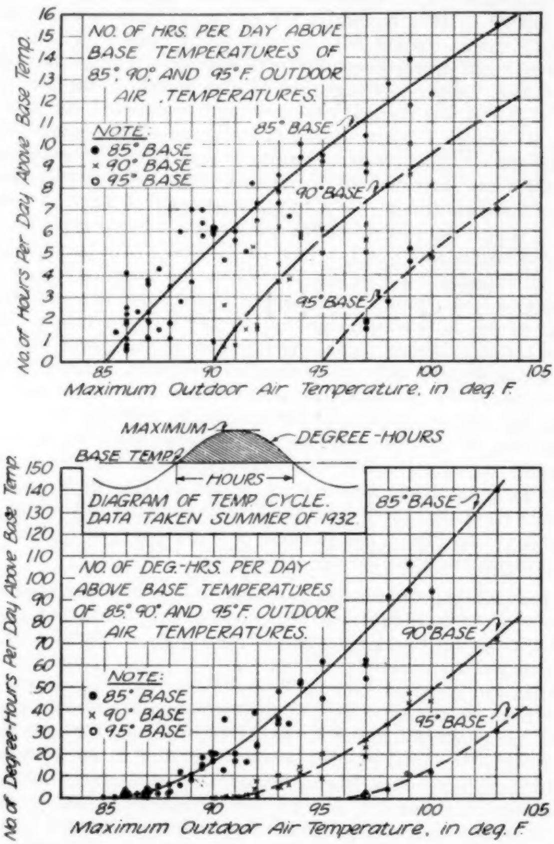


FIG. 11. CURVES SHOWING RELATION BETWEEN MAXIMUM OUTDOOR AIR TEMPERATURE AND NUMBER OF DEGREE HOURS AND NUMBER OF HOURS PER DAY ABOVE BASE TEMPERATURE

in the same locality. During the ten years, it varied from a maximum of 2,502 degree-hours to a minimum of 327 degree-hours. The latter represents only about one-eighth of the maximum. Futhermore, during this ten-year period there were four years in which the number of degree-hours was less than one-fourth of the maximum, six years in which it was less than one-third and eight years in which it was less than two-thirds of the maximum. That

the cycle represented by the ten years selected presents nothing unusual in the way of variation is shown by Fig. 12, in which the average of all the maximum temperatures for each day in the months of June, July, August, and September are plotted against the years, over a period of 45 years. It may be

TABLE 3. NUMBER OF DAYS OF STATED MAXIMUM TEMPERATURE AND NUMBER OF DEGREE HOURS ABOVE 85 F FOR SEASON FROM JUNE 1 TO OCTOBER 1 AT URBANA, ILL.

Range of Maximum Outdoor Temperature Deg F	Mean of Range	No. of Deg Hours above 85 F per Day (from Fig. 11 at Mean of Range	1932*		1931		1930		1929		1928	
			No. of Days	Total Deg Hours	No. of Days	Total Deg Hours	No. of Days	Total Deg Hours	No. of Days	Total Deg Hours	No. of Days	Total Deg Hours
85– 87	86	1.0	23	23	22	22	15	15	17	17	19	19
88– 90	89	10.5	16	168	14	147	13	137	15	158	14	147
91– 93	92	30.5	11	336	9	274	15	457	5	152	8	244
94– 96	95	56.0	4	224	18	1008	7	392	0	0	0	0
97– 99	98	86.0	6	516	7	602	7	602	0	0	0	0
100–102	101	119.0	1	119	0	0	5	595	0	0	0	0
103–105	104	152.0	1	152	0	0	2	304	0	0	0	0
Total................			62	1538	70	2053	64	2502	37	327	41	410

Range of Maximum Outdoor Temperature Deg F	Mean of Range	No. of Deg Hours above 85 F per Day (from Fig. 11 at Mean of Range)	1927		1926		1925		1924		1923	
			No. of Days	Total Deg Hours	No. of Days	Total Deg Hours	No. of Days	Total Deg Hours	No. of Days	Total Deg Hours	No. of Days	Total Deg Hours
85– 87	86	1.0	12	12	14	14	18	18	12	12	14	14
88– 90	89	10.5	10	105	17	179	20	210	12	126	12	126
91– 93	92	30.5	6	183	8	244	17	518	4	122	7	213
94– 96	95	56.0	4	224	6	336	8	448	3	168	6	336
97– 99	98	86.0	0	0	0	0	3	258	0	0	1	86
100–102	101	119.0	0	0	0	0	1	119	0	0	0	0
103–105	104	152.0	0	0	0	0	0	0	0	0	0	0
Total................			32	524	45	773	67	1571	31	428	40	775

noted also from Fig. 12, that the shape of the degree-hour curve, plotted from the date in Table 3, closely approximates the shape of the average maximum temperature curve. Hence, the variations shown by the 10-year cycle selected may be regarded as typical of the variations that might occur in any 10-year cycle during the whole period of 45 years, and both the curves and the table serve to emphasize the futility of any attempt to calculate the probable cooling load for a given season from a mean value of the number of degree-hours based on the average for a cycle of seasons.

DAILY VARIATION IN OUTDOOR TEMPERATURES

It may be noted from the outdoor temperature curves shown in Figs. 6 and 7, that the difference in temperature between the maximum in the daytime and

the minimum during the preceding night ranged from 19 to 28 deg, and that for a period of six hours, or more, the outdoor temperature was at least 20 deg below the maximum attained during the same day. An analysis of all of the daily temperatures during the months from June 1 to October 1 proved that, with very few exceptions, a difference of at least 20 deg was exhibited. The exceptions occurred when the maximum was 85 deg or less. The average difference when the maximum was from 75 to 85 F was 19 deg. For maximums between 85 and 95 F, the average difference was 23 deg, and for maximums above 95 F it was 25 deg. This indicated that for considerable periods at night the outdoor air constituted a reservoir from which air could be drawn in order to cool the residence at night, and reduce the cooling load required during the following day. For a large portion of the time, enough cooling could probably

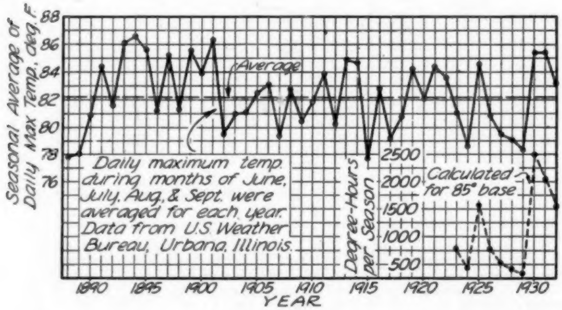


FIG. 12. GRAPHICAL RECORD OF SUMMER TEMPERATURES IN URBANA, ILL., DURING YEARS 1888 TO 1932

be effected in this way to make it unnecessary to start the cooling plant the next day, particularly in the case of an insulated structure. It should be emphasized, however, that in order for this method to be effective, a fan large enough to circulate sufficient outdoor air to cool the walls and contents of the structure, and not merely to cool the air inside of the building, would be required. Just what could be accomplished in this respect was not determined by this investigation, but the results indicate that this would be a fruitful field for future investigation.

RESULTS OF TESTS ON UNIT COOLERS

The data obtained from the unit coolers were too meager to permit any conclusions in regard to allocation of the cooling load. Some data were obtained on the operation of these units, however, and may prove of interest. On continuous operation, one unit cooled three rooms to a dry-bulb temperature of 76 F when the outdoors temperature was 85 F. The relative humidity at this time was 55 per cent outdoors and 59 per cent indoors. The ice meltage was 36.1 lb per hour, and the air passing through the unit was cooled from a dry-bulb temperature of 76 F to one of 65 F. The dehumidification amounted to about 0.25 lb per hour and the wet-bulb temperature of the entering air was

66 F and of the leaving air was 61 F, or a reduction of 5 deg in the wet-bulb temperature.

On intermittent operation, this unit cooled one room to a dry-bulb temperature of 75 F when the outdoor dry-bulb temperature was 81 F. Both outdoor and indoor relative humidities were 55 per cent. The overall ice meltage was 20.1 lb per hour. The meltage on the standby periods was 6.1 lb per hour and during the actual running periods was 31.5 lb per hour. The time of actual running was 44.5 per cent of the total time. In this case, the unit reduced the dry-bulb temperature of the air passing through it from 75 F to 60.5 F. The wet-bulb temperature of the air passing through the unit was reduced from 63.5 F to 55.2 F, or 8.3 deg. The dehumidification was approximately 0.25 lb per hour.

Conclusions

The following conclusions may be drawn, subject to the limitations of the conditions under which the tests were run:

1. The ducts and registers of a central forced-air heating system can be successfully adapted as a distributing system for cooled air in the summertime without material alterations.

2. There is no tendency for a pool of cold air to collect near the floor if the temperature of the air entering the room through the registers is not lower than 60 F.

3. Baseboard registers can be used without objectionable drafts resulting if the velocity of the air at the register face is below 300 fpm although wall registers located 7 ft above the floor are preferable to baseboard registers.

4. A building of the type of the Research Residence may require the equivalent of two tons of ice in 24 hours including the basement load on days when the maximum outdoor temperature reaches 100 F if an effective temperature of approximately 72 deg is maintained indoors.

5. An effective temperature somewhat higher than 72 deg may be satisfactory for residence service, in which case more conservative results for the cooling load can be obtained.

6. The use of awnings at all windows in east, south, and west exposures may result in savings of from 20 to 30 per cent in the required cooling load.

7. The cooling load per degree difference in temperature is not constant, but increases as the outdoor temperature increases.

8. The heat lag of the building complicates the estimation of the cooling load under any specified conditions and makes such estimates, based on the usual methods of computation, of doubtful value.

9. The seasonal cooling requirements are extremely variable from year to year, and the ratio of the degree-hours exhibited by any two seasons occurring within a ten-year period may be as high as 7.5 to 1. Hence an average value for the degree-hours per season is comparatively meaningless.

10. The results of these tests suggest two phases, in regard to which further investigation is desirable. The first concerns the load requirements when unit

coolers are used in one or more partially isolated rooms with the rest of the building not cooled. The second concerns the use of a fan at night either to provide more comfortable conditions during the following day without provision for cooling, or to reduce the load required for cooling during the following day.

DISCUSSION

M. G. HARBULA: I should like to ask why a return air by-pass around the cooling coils was used.

PROF. A. P. KRATZ: A return air by-pass was used because by using this arrangement as a temperature control, a continuously operating plant could be obtained, and it was desired to avoid the uncertainties and inaccuracies in measurements that would have resulted from an intermittently operated plant.

MR. HARBULA: If you had been familiar with another method of temperature control, would the by-pass have been used?

PROFESSOR KRATZ: We were familiar with other methods of temperature control, but selected the by-pass method because it offered the advantage of continuous operation.

MR. HARBULA: Did the use of a by-pass increase or decrease the amount of refrigeration used, which in effect would mean an increase or decrease in the amount of ice being melted to produce refrigeration?

PROFESSOR KRATZ: We have no data to indicate whether the use of the by-pass would increase or decrease the amount of ice meltage required to cool the house, but have no reason to believe that it would do either when used merely as a method of controlling the room temperature.

MR. HARBULA: The anticipation of this discussion before a meeting of the A. S. H. V. E. has been my main reason for attending this Annual Meeting, because of the controversy in regard to the use of the by-pass, and because THE A. S. H. V. E. GUIDE 1933 mentions that more refrigeration would be used on a job if a by-pass was not installed in connection with an air conditioning system.

A. C. WILLARD: The results presented in this paper required the determination by the research staff of a truly enormous amount of experimental data. I think I may safely say that, of all the research problems in the field of heating and ventilation which we have undertaken at the University of Illinois, this problem of the summer cooling of an actual residence has required the most intensive effort in the collection and compilation of data and their subsequent analysis which we have ever given to any problem.

You have seen merely a cross section of the data and results of these summer cooling tests in the paper which has been presented, and I, therefore, wish to extend an invitation to the members and guests of the Society to visit Urbana and inspect any phase of this summer cooling work which may be of particular personal interest to you. You will find the research staff ready and willing to put their time against yours in a discussion of either the tests themselves or the results and conclusions drawn from the tests, as presented here today. This invitation to inspect our research work includes, also, any or all of our other research activities in the field of heating and ventilation now in progress at the University of Illinois.

No. 948

SUMMER COOLING OPERATING RESULTS IN A DETROIT RESIDENCE

By J. H. WALKER † (*MEMBER*) AND G. B. HELMRICH ‡ (*NON-MEMBER*)
DETROIT, MICH.

This paper is presented in collaboration with the paper entitled, Study of Summer Cooling in the Research Residence at the University of Illinois, by A. P. Kratz and S. Konzo

THIS paper gives the results obtained from the study of a residence cooling system under normal conditions of occupancy and with only a moderate degree of cooling as compared with the conditions maintained in the Research Residence at the University of Illinois.[1] It was particularly desired to determine the minimum requirements for producing a moderate improvement in comfort and to determine the feasibility of cooling an entire residence with a cooling system requiring only a small investment. The work was undertaken by The Detroit Edison Company.

There is some difference of opinion as to the extent to which artificial cooling of residences is economically feasible. Some believe that the cooling of a first floor living room and one bedroom is all that should be attempted, because of the high cost of installing a complete system or because of the operating cost. It is true perhaps that if it is necessary to install a system of pipes or air ducts in an existing house the installation cost may restrict the number of rooms which may be cooled. In a house which has a warm air heating system, however, the existing ducts offer an excellent means of distributing the cooled air, and thus there is a large number of homes in which the application of summer cooling is a relatively simple problem. The operating cost is reasonable if only a moderate amount of cooling is done, at least in a normal summer in the vicinity of Detroit, Michigan; and a moderate amount of cooling throughout the entire house is considered preferable to concentrating the effect in two or three rooms.

It is obvious that both the economics of residence cooling and the design of cooling systems will vary widely in different parts of the country and general conclusions should not be drawn from the results in this one installation.

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† Superintendent of Central Heating, The Detroit Edison Company.
‡ Engineer, Construction Bureau, The Detroit Edison Company.
[1] Study of Summer Cooling in the Research Residence at the University of Illinois, by A. P. Kratz and S. Konzo, A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933.

Presented at the 39th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Cincinnati, Ohio, January, 1933, by J. H. Walker.

The results should be indicative, however, of what may be expected from installations of the same type, operating under similar conditions.

THE RESIDENCE

The residence in which the tests were made is located in Birmingham, a suburb of Detroit. It is of moderate size and of substantial frame construction. The heating system is a mechanical warm air system having return ducts from all of the rooms.

The cooling was done by means of ice which was chosen because it was believed that the ice cooling system is appropriate for this kind of service. A system using ice is simple and inexpensive to install. The cost of ice is rather high compared to the cost of operating an individual refrigerating machine, but the operating season is of short duration. Ice also possesses the merit of flexibility as to the rate of refrigeration. By the simple expedient of varying the melting rate the cooling effect can be changed through a wide range.

FIG. 1. THE DETROIT RESIDENCE IS A TYPICAL WELL-BUILT, FRAME HOUSE OF MODERATE SIZE

Note: Dimensions 46 ft by 26 ft. Eight rooms cooled—4 on each floor. Total space cooled 17,200 cu ft. All openings weatherstripped. Walls built up of shingle siding, 3 ply waterproof building paper, pine sheathing, studding, lath and plaster, with flexible insulation nailed between studding. Attic floored over with flexible insulation between attic floor joists. Shutters or awnings on all windows exposed to sun. Winter heat loss 93,000 Btu per hour for 70 F inside and 0 deg F outside.

The only additional equipment which was necessary to provide for complete cooling in this residence was a tank for the ice, a coil placed in the air circuit, and a pump to circulate the ice water through the coil. Actually other features were added for experimental purposes but the tank, coil, and pump are the only essential elements, and the experience gained indicates that all of these could have been smaller and more simply arranged. The actual installation cost was comparatively high because of the necessity of providing a new fan and

rearranging the ducts, but it is estimated that, had the cooling equipment been installed at the time the house was built, the added cost would not have exceeded $500.00, and might have been as low as $300.00.

The arrangement of the equipment is shown in Fig. 2. The ice tank is buried, without insulation, just outside the basement wall. It is charged through a trap door which is close to the driveway and therefore readily accessible to the delivery truck. The cooling coil is a standard automobile radiator. The pump is an inexpensive centrifugal pump, mounted on rubber discs and connected with the piping by rubber hose. The fan is a standard design of

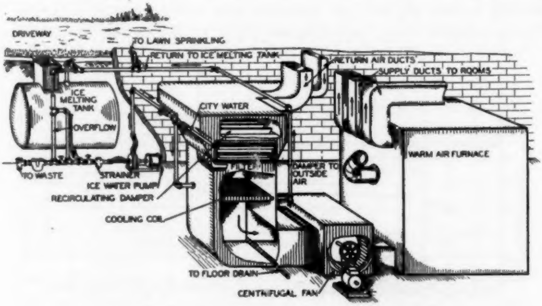


Fig. 2. The Cooling Apparatus

Notes: Furnace Fan—Centrifugal type, rated at 2,600 cfm at ⅜ in. static pressure. Outlet velocity 1,250 fpm at a speed of 482 rpm. Motor ½ hp capacitor type.
Ice Tank—5 ft 7 in long by 4 ft diameter.
Cooling Coils—Standard automobile radiator having 492 sq ft of surface.
Pump—Centrifugal pump rated at 10 gpm against a total head of 35 ft. Driven by ½ hp capacitor-type motor.
Note that the required sizes and capacities of these units in most cases proved to be considerably smaller than actually installed. (See text.)

double inlet centrifugal furnace fan and is set on strips of soft rubber, while the driving motor is set on cork to insure quietness of operation.

Provision for Outside Air

At all times when artificial cooling was done the system was operated with complete recirculation, no outdoor air being used. Provision was made, however, to draw in outside air when desired and this feature proved to be extremely valuable, and should be considered an essential part of such a system. In Detroit, as in most parts of the country, there are many nights and a few days during the summer when the outside air is cooler than the inside air, but there is no breeze to bring it into the rooms. The connection for bringing in outside air was simple to provide and satisfied all cooling requirements at certain times for merely the slight cost of the fan power.

Cooling Is a Cyclic Process

Engineers are inclined to regard cooling problems as the reverse of, but similar to, heating problems and to consider the process of cooling as that of

maintaining the interior of a building at a constant temperature and absorbing heat which flows inward much as heat flows outward in winter. There are several essential differences between the heating and cooling processes however. One difference is that there is a wide range of temperatures through which people are reasonably comfortable in summer, whereas in winter they are accustomed to having indoor temperatures controlled within rather close limits.

Another difference concerns a point already mentioned, that in a large part of the country the conditions which exist out-of-doors during many days, and especially during many nights, in summer are entirely comfortable and all that is required is to create those conditions indoors by drawing in outside air; no

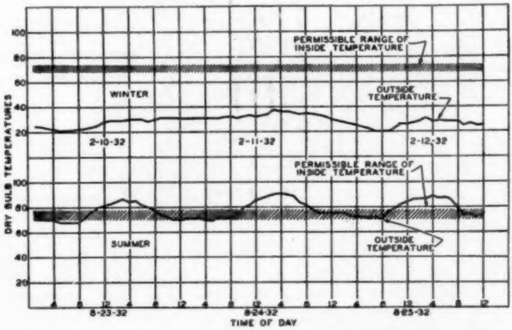


FIG. 3. RELATION OF OUTDOOR TEMPERATURES TO DESIRABLE INDOOR TEMPERATURES FOR WINTER AND SUMMER CONDITIONS

artificial cooling is required. Thus while heating is required continuously in winter, summer cooling is necessarily intermittent and to a great extent cyclic. This is well illustrated by Fig. 3 which shows the relation between outdoor temperatures and acceptable indoor temperatures in summer and in winter. There are of course occasional periods in Detroit and elsewhere when the night temperatures are high; but there were none in Detroit during the past summer and only a very few in the hot summer of 1931.

TEMPERATURE RISE WITHOUT COOLING

In view of the cyclic nature of the cooling process and the well-known lag in the temperature rise in a tight, well-shaded house, the question naturally arose as to what the indoor temperature would be if no artificial cooling were done. This house was well protected from the sun, being equipped with either awnings or shutters on all windows exposed to direct sunlight; and the walls were well insulated. How effective these provisions are is well illustrated in Fig. 4. Although the outside temperature during this test was not extremely high the slow rise of the inside temperature is quite evident. The average rate

of rise of the first floor temperature was only about ½ deg per hour for a differential between inside and outside of approximately 10 deg.

Window shading, insulation, and thermal capacity of the structure are only partially effective in keeping the house cool, however, and could not be considered to obviate the need for artificial cooling. Eventually, during a protracted hot spell the inside temperature would become uncomfortably high.

METHOD OF OPERATION OF THE COOLING SYSTEM

The fan was operated experimentally at various speeds but at the most satisfactory point it delivered approximately 2,110 cfm, equivalent to an aver-

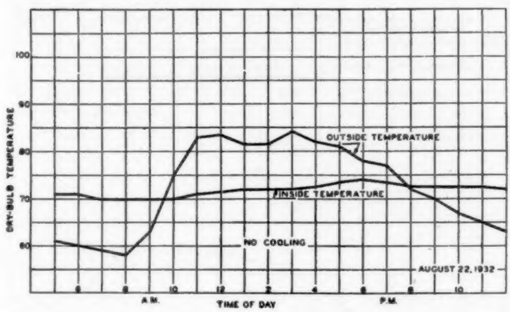


FIG. 4. INSIDE TEMPERATURE WHEN NO COOLING WAS DONE

age air change of over seven times per hour for the entire space cooled. The temperature of the air leaving the registers was from 59 to 68 F, averaging about 65 F during the cooling period. The velocity varied from 100 to 317 fpm.

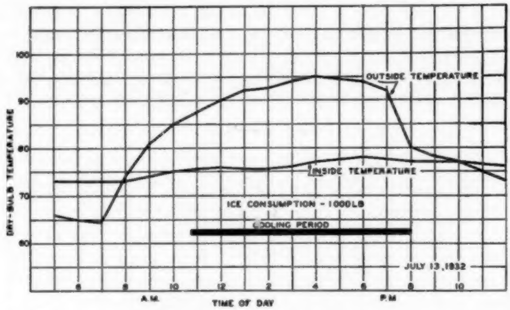
The temperature of the water entering the coil was 47 F which was low enough to cause a considerable amount of moisture to be condensed from the air. The indoor humidity ranged from 45 to 60 per cent while the system was in operation which seemed to be entirely satisfactory, there being no noticeable feeling of clamminess at any time. The dehumidifying load in a residence is of course normally small compared to the cooling load except for occasional temporary loads caused by cooking and bathing.

The air supply registers were all located just above the baseboard. There were no seriously uncomfortable drafts except where the arrangement of the furniture was such that the occupants found themselves in the immediate vicinity of the registers. It would have been desirable to have had deflectors in front of a few registers but balancing the system by adjusting dampers quite effectively moderated objectionable drafts. There seemed to be no serious temperature stratification of the air in any of the rooms and no noticeable short-circuiting of the air to the outlet grilles. The temperature difference

between the floor level and the breathing line, measured on different occasions, averaged about 4 deg.

Results Obtained

During the early part of the summer the system was operated freely and without particular regard to economy and the ice consumption, on days when the outside temperature reached 95 F, varied from 1,000 to 2,500 lb. Indoor



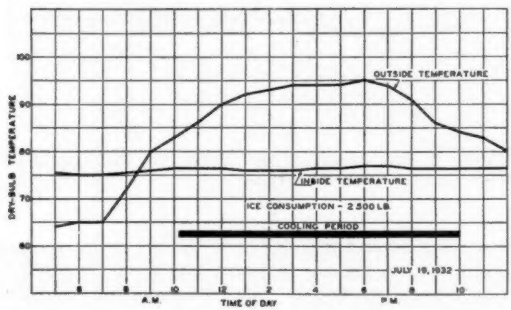


Fig. 5. Results with Unrestricted Use of the Cooling System

temperatures of 75 F to 77 F were held continuously throughout the day and while these conditions seem ideal to a continuous occupant, to a person entering or leaving there was a greater temperature differential than was entirely comfortable. The cost of operation with ice at $5.00 per ton with this unrestricted cooling was of course rather high.

The results obtained on two typical days with this type of operation are shown in Fig. 5. The ice consumption was 1,000 lb on one day and 2,500 lb on the other day with inside temperatures maintained at not above 78 F and 77 F respectively.

It soon became apparent that there was a very wide range of temperatures that could be maintained in the house which could be considered as satisfactory, and it was further evident that to maintain temperatures as low as 76 F and

78 F on the hottest days required an ice consumption that was out of all proportion to the slightly increased comfort obtained as compared with a slightly higher temperature. Therefore during the month of August a more moderate amount of cooling was done and the ice consumption was thereby greatly reduced.

The rate of ice consumption was limited by covering part of the coil surface with a sheet metal baffle and by throttling the flow of water through the coil. The ice was not delivered each day until afternoon and then in a limited quantity of 500 to 700 lb. Thus a moderate cooling effect was obtained. The temperature of the house rose gradually as the outside temperature rose, but when the cooling system was put in operation by starting the pump, the

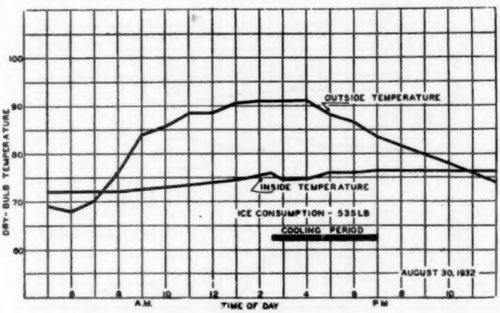


FIG. 6. INSIDE AND OUTSIDE DRY-BULB TEMPERATURE FOR AUGUST 30, 1932

Note that a satisfactory condition was obtained on this day with a consumption of only 535 lb of ice

rise of the house temperature was checked. This method tended to give what is probably the best temperature regulation, that is, an increasing differential between inside and outside temperature as the outside temperature increases.

One feature was noted in these tests that is usually true—it is impracticable to attempt to hold a large *differential* between outdoor and indoor temperatures when the outdoor temperature is falling. It would be practically impossible to bring down the inside temperature at a rate commensurate with a rapidly falling outside temperature.

With the type of system described, the proper method of operation is to stop the flow of water through the coil when the outside temperature is falling in the late afternoon but to continue to circulate the air with the fan. When the outside effective temperature becomes less than that inside, the outside-air damper should be opened and outside air drawn in.

The results obtained with this moderate amount of cooling are shown in Figs. 6 and 7. They are highly important as illustrating what can be done with a reasonable consumption of ice. The conditions were entirely satisfactory and probably much better physiologically than a lower temperature

would have been. The ice consumption on this day was 535 lb and on other similar days it varied from 500 to 700 lb. These experiments thoroughly demonstrated the important fact that by the use of a moderate amount of ice a satisfactory degree of comfort could be obtained. Table 1 shows the complete operating data for the day represented in Figs. 6 and 7.

An incidental feature of the installation was a provision for using city water instead of ice as the cooling medium. City water, if its temperature is low enough, is an economical medium for cooling. In this case the water came from deep wells and had a temperature of 58 F during June, which rose to about 61 F in September. A few tests were made which indicated that on

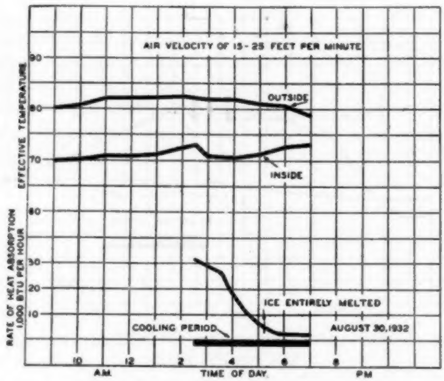


FIG. 7. INSIDE AND OUTSIDE EFFECTIVE TEMPERATURES FOR THE SAME DAY ILLUSTRATED IN FIG. 6

days which were not extremely hot, a considerable amount of cooling could be done in this way. The piping was so arranged that the water used for cooling could then be utilized for lawn sprinkling.

The use of city water appears to have only a limited possibility of application, however. In many localities its temperature is unsuitable and in others there would undoubtedly be restrictions placed on its use for this purpose; but where cold water is available it is a relatively economical source of cooling. Comparing, for example, the cost of operation with ice, city water, and an electrically-operated compression refrigerating machine, the figures of Table 2 were obtained, using the prevailing local unit costs for ice, water and electricity.

OVERALL RESULTS FOR ENTIRE SUMMER

The method of operation of the system was varied considerably during the summer and the total ice consumption is, therefore, not representative of the

best operating method, being somewhat higher than necessary for a moderate amount of cooling. However, the actual summer's results, as given in Table 3, are interesting and show a surprisingly reasonable operating cost.

TEMPERATURE CONTROL

The question of temperature control naturally arose and required some experimentation. A thermostat, maintaining a constant temperature, was obviously unsuitable because the inside temperature should be permitted to rise when the outside temperature rises. The simple method finally adopted was to adjust the system for a moderate rate of heat absorption or, in other words, a moderate ice melting rate, and to operate continuously for a few hours each day, permitting the temperature to rise as it would. Control is thus obtained by regulating the number of hours of operation per day. The system was put into operation by merely starting the fan and pump by means of conveniently located switches and was operated until the outside temperature fell in the afternoon or until the ice was exhausted. The cost of operation was controlled by putting into the ice tank each day only the amount of ice found to be sufficient for a moderate amount of cooling.

TABLE 1. OPERATING DATA FOR A DAY WITH MODERATE COOLING, AUGUST 30, 1932

Time of cooling	2:30 P.M.–7:00 P.M.	
Pounds of ice used	535	
Ice melting time—hours	2.75	
Ice melting rate—pounds per hour	195	
Pounds of water condensed from air	12.87	
Inside Temperatures—First Floor	*For 24 Hours*	*For Cooling Period*
Maximum	77.2 F	77.2 F
Average	74.8 F	76 F
Average wet bulb	......	66.8 F
Outside Temperatures		
Maximum	91 F	91 F
Average	80 F	88 F
Average wet bulb	......	74.8 F
Average temperature of air to coil	71.2 F	
Average temperature of air from coil	64.8 F	
Average temperature of air from grilles	65.2 F	
Average velocity of air from grilles—fpm	190	
Air circulated—cfm	2,110	
Air changes per hour	7.3	
Average power input to motors—kw	0.84	
Total kwhr consumed	3.78	
Minimum temperature of water to coil	48.5 F	
Minimum temperature of water from coil	55.0 F	
Quantity of water circulated—gpm	4.93	

RELATION OF COOLING REQUIREMENTS TO DEGREE-HOURS

The factors which affect cooling load and ice consumption are very complex. In general the cooling and dehumidifying load has two components—the external load which is largely a function of outside temperature and the load which is due to internal sources and is nearly independent of outdoor conditions. In a residence the internal load is usually negligible.

It would be convenient to have an approximate rule for determining the ice or current consumption, similar to the degree-day method of calculating fuel consumption for heating. It was noted that the cooling system was operated, using ice, on those days when the outside temperature at some time during the day exceeded 85 F. If therefore 85 F is arbitrarily taken as the base temperature and the degree-hours above 85 F computed, the following comparison with the ice consumption is obtained:

	Degree-Hours above 85 F	Ice Consumption (Lb)	Per Cent of Total Degree-Hours	Per Cent of Total Ice Consumption
July	339	8,950	67	65
August	164	4,782	33	35
Total	503	13,732	100	100

This fairly close agreement between degree-hours and ice consumption, as expressed in per cent, may be mere coincidence or may indicate that the method used is a simple yard stick for determining cooling requirements at various temperatures. It is interesting to note the figures for preceding years for degree-hours above 85 F in Detroit, as follows:

Year	Total Degree-Hours above 85 F
1932	529
1931	1,341
1930	935
1929	490
1928	260
1927	334
1926	351
1925	521
1924	224
1923	402
Ten year average	538.7

MISCELLANEOUS NOTES ON OPERATION

The capacity of some of the items of equipment was found to be considerably in excess of the required capacity. This was true of the pump, which was finally throttled down to a flow of 6 gpm. The ice tank could also have been smaller if daily ice deliveries are assumed. A capacity of 45 cu ft would have

been ample. The coil, too, proved to be about 50 per cent larger than necessary for the ice melting rate finally used.

Experiments in which some of the air was by-passed around the cooling coil indicated that such by-passing was unnecessary. The essential purpose of a by-pass would be to permit the mixing of some relatively warm recirculated air with the air which is cooled to a low temperature for dehumidification, so that the resultant temperature of introduction into the rooms would not be uncomfortably low. It was found, however, that with a water temperature of 47 F sufficient dehumidification was obtained for comfort and the air temperature at the registers was 62 F which was not uncomfortably low.

A feature that would probably further reduce the operating cost would be to cool only the first floor in the day time and to cool the second floor in the evening. In new installations the duct systems could be so designed that only one or two dampers would require changing in order to divert the cooling effect from one floor to the other.

TABLE 2. CALCULATED COMPARATIVE OPERATING COSTS WITH DIFFERENT METHODS OF COOLING

	Ice	City Water (Well Water)	Electricity (In a 2-ton Compression Unit)
Unit cost	$5.00 per ton	$0.20 per M gal	$0.0225 per kwhr
Heat absorbing capacity	344,000 Btu per ton[a]	50,000 Btu per M gal Initial temp 58 F Discharge temp 64 F	10,236 Btu per kwhr[b]
Cost per M Btu of cooling effect	1.45 cents	0.40 cents	0.29 cents[c]

[a] Based on ice water temperatures rising to 60 F at end of run (all water recirculated).
[b] Equivalent to 33.5 kwhr to produce the same cooling effect as a ton of melting ice—coefficient of performance of compression unit assumed to be 3.
[c] Includes 0.07 cents for condensing water.

The arrangement of the ice tank was quite satisfactory. The standby loss was about 1,600 Btu per hour or 10 lb of ice. It was found desirable to use only a small amount of water in the bottom of the tank because there is an appreciable irrecoverable loss if a large volume of water is cooled. An inside tank could be used but should be provided with a charging chute from outside and should be insulated against noise and excessive heat transfer.

The matter of ice delivery will demand study on the part of the ice companies when this method of cooling becomes more common. Deliveries must be prompt because it is sometimes difficult to predict more than three or four hours in advance when ice will be needed. Some method of scheduling deliveries according to the outdoor temperature may be desirable.

CONCLUSIONS

1. The results of this investigation lead to the definite conclusion that in a well-shaded and insulated house, in the Detroit, Michigan, climate, summer

cooling with ice is economically feasible provided a moderate amount of cooling is done.

2. A well designed mechanical warm air heating system is readily adapted to summer cooling and the additional equipment required for cooling is not extremely costly.

3. The use of outside air at such times as its temperature is below the

TABLE 3. OPERATING DATA FOR COOLING SEASON, 1932

Total number of days when artificial cooling was used	22
Total number of hours when artificial cooling was used	135
Total equivalent amount of ice used*	18,166 lb
Average ice consumption per day of artificial cooling	826 lb
Average ice consumption per hour of actual operation	135 lb
Temperature, during days of artificial cooling	
Maximum inside temperature at any time	80 F
Average inside temperature	74.7 F
Maximum outside temperature	95 F
Average outside temperature for 24 hours	77.8 F
Average outside temperature for cooling periods	83.7 F
Average dry bulb temperature difference between indoors and outdoors during hours of operation	8.1 F
Average effective temperature difference between indoors and outdoors during hours of operation	6.2 F
Power Use	
Electricity used for pump and fan, entire season	94 kwhr
Electricity used per ton of ice melted	10.4 kwhr
Approximate amount of electricity used to manufacture the ice, at 50 kwhr per ton	454 kwhr
Estimated amount of electricity which would have been used in an individual refrigerating machine to produce the same cooling effect	304 kwhr
Total electricity used (94+454)	548 kwhr
Total estimated use of electricity, assuming individual refrigerating plant (94+304)	398
Costs	
Ice (including equivalent of city water) 9.08 tons at $5.00 per ton	$45.40
Electricity 94 kwhr at 2¼ cents per kwhr	$ 2.12
Total operating cost	$47.52
Average total cost per day when cooling was used	$ 2.16
Average cost per hour of actual operation	$ 0.352

* *i.e.* Including an amount of ice equivalent to the cooling effect of the city water which was used on certain days.

inside temperature is desirable but does not, by itself, give adequate comfort in extremely hot weather.

DISCUSSION

J. H. WALKER: We have attempted to reconcile the ice consumption of the Research Residence with that of the Detroit residence. Figure A shows a comparison of the characteristics of the two houses. You will notice that the Detroit residence is larger but that, largely because of its better insulation and the fact that it is equipped with weatherstrips, both the computed winter heat loss and the computed

summer heat gain are lower than for the Research Residence. With our present knowledge of the subject the figures of heat gain are probably not very accurate.

There was a great difference in the outside temperature conditions in the two cities.

Table A. Comparison of Ice Consumptions

	Detroit Residence	Research Residence
Space cooled—cubic feet	17,200.00	14,170.0
Computed Winter Heat Loss, Btu/hour (70 F differential)	93,000.00	119,000.0
Computed Heat Gain Btu/hour (20 F differential)	32,389.00	33,500.0
Degree Hours above 85 F for Entire Season	529.00	1,471.0
Ice consumption for season—tons	9.08	43.3

Correcting ice consumption of Research Residence to Detroit conditions

$$43.3 \text{ tons} \times \frac{32{,}389 \text{ Btu}}{33{,}500 \text{ Btu}} \times \frac{529 \text{ degree-hours}}{1471 \text{ degree-hours}} = 15.03 \text{ tons equivalent ice consumption.}$$

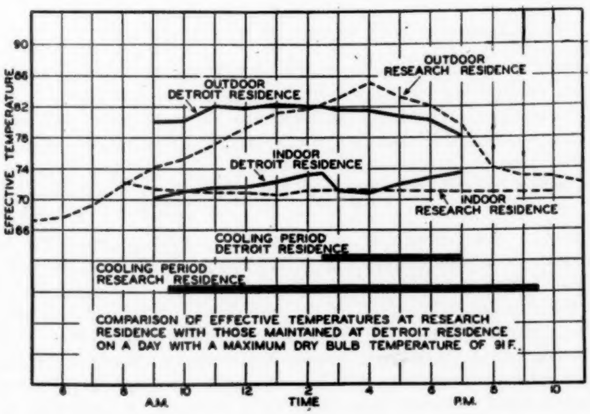


Fig. A. A Comparison of 2 Typical Days of Operation in the Detroit and Research Residences

The number of degree-hours above 85 F was 1471 for Urbana and 529 for Detroit, a ratio of 2.78 to 1, and this accounts for a large part of the difference between the Urbana ice consumption of 43.3 tons and the Detroit consumption of 9.08 tons. The number of degree-hours above 85 F is obviously not a true index of cooling requirements, but it is the best index that we have as yet.

If, remembering these reservations, we multiply the Urbana ice consumption by the ratio of the heat gains and by the ratio of the degree hours, we obtain a figure which

represents approximately the Urbana ice consumption corrected to Detroit conditions. This figure works out (see Table A) to be 15.03 tons. The difference between 15.03 tons and 9.08 tons, the actual Detroit consumption, is probably due to the greater amount of cooling done in the Research Residence and to the fact that no advantage was taken of the potential cooling effect of the outside air as was done in the Detroit residence.

Fig. A, which compares 2 typical days' operation for the 2 houses, illustrates the differences in operating methods. The outside conditions are not exactly similar, although the outdoor dry bulb temperature in both cases attained a maximum for the day of 91 F. The greater amount of cooling and the longer period of ice cooling in the Research Residence are apparent.

It is therefore probable that a considerably less amount of ice could be used in the Research Residence to give acceptable results; economy of ice consumption was not sought in last summer's work. We also believe that the consumption in the Detroit residence can be reduced in view of our experience.

No. 949

CORROSION AS RELATED TO AIR CONDITIONING EQUIPMENT

By R. M. PALMER *(NON-MEMBER), NEW YORK, N. Y.

IT IS customary in the air conditioning industry to emphasize the factor of temperature and clarity of a given water and permit these properties to serve as a basis for adopting the use of any given available supply. Such a procedure is quite inadequate and has many times given rise to serious operating difficulties. A clear water having a satisfactorily low temperature throughout the year may prove to be an extremely dangerous water to use in air-conditioning equipment.

PROPERTIES OF WATER TO BE CONSIDERED

What are the other properties of water which should be investigated before adopting a given water as a source of supply for condenser water and for spray chamber water for air conditioning systems? In general, a chemical analysis of the water should be made. It may not be necessary to make a complete chemical analysis, but it certainly is necessary to determine the total alkalinity, the total acidity, the total hardness and the *pH* values of a water before adopting it for use.

What will these analyses disclose? It is known that a highly alkaline water, if the alkalinity is due to calcium and magnesium carbonates, will be a scale-forming water. Scale may or may not develop in the cold. Calcium carbonate has a reverse solubility curve. That is, its solubility in water decreases as the temperature of the water increases. Because of this fact it is possible at times to use waters having a relatively high calcium carbonate content for spray washer purposes and for condenser cooling purposes. It must be appreciated, however, that scale may form at normal temperatures. The interior surface of a sample of piping taken from cold supply line of an office building in Buffalo, New York, had a deposit of the carbonates of calcium and magnesium. In this case the Niagara River supply, which flowed through this pipe, contains sufficient amounts of the carbonates of calcium and magnesium to bring about deposition of the scale in question, in the cold. The rate of scale deposition will depend not alone on the temperature factor but will also depend on velocity of flow.

Soft waters, or waters having a low calcium and magnesium carbonate

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* Vice-President, Ferro-Nil Corporation.
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alkalinity, are invariably corrosive. The corrosive action of such waters may be caused by the presence of dissolved oxygen, dissolved carbon dioxide or dissolved acids of organic or inorganic origin.

pH-VALUE AND HYDROGEN-ION CONCENTRATION

The determination of the *pH* of a water discloses certain information about what may be expected as to its corrosive properties. Pure distilled water will conduct an electric current to a very slight degree. This shows that a very small proportion of the water is dissociated into *H* and *OH* ions. By the mass law,

$$\frac{\text{Concentration of } H \text{ ions} \times \text{Concentration of } OH \text{ ions}}{\text{Concentration of Undissociated } H_2O} = \text{a constant} \qquad (1)$$

Since the amount of undissociated water is relatively large, it can be taken as a constant, and Equation 1 therefore becomes

$$\text{Concentration of } H \text{ ions} \times \text{Concentration of } OH \text{ ions} = \text{a constant} \qquad (2)$$

By electrical conductivity measurements, this constant has been found to be 1/100,000,000,000,000 or 10^{-14} at 22 C. Since in pure distilled water the number of *H* ions is equal to the number of *OH* ions, each must have a concentration of 1/10,000,000 or 10^{-7}.

This method of expressing *H*-ion concentration is very inconvenient and clumsy. Sorensen, therefore, suggested the use of the term *pH*, the *pH* value being the logarithm of the reciprocal of the hydrogen-ion concentration. Thus,

$$pH = \log \frac{1}{(H^+)} \qquad (3)$$

Just as an acid solution is said to be normal when it contains 1 gram of ionizable hydrogren per liter, so a solution is said to be normal with respect to hydrogen ions when it contains 1 gram of ionized hydrogen per liter. Since $\frac{1}{(H^+)}$ is the reciprocal of the normality of *H* ions in a solution, the *pH* value can also be defined as the logarithm of the denominator expressing the normality of *H* ions such as *N*/10, *N*/100, *N*/1000, etc.

This method of expressing *H*-ion concentration has now been generally adopted. In the case of pure distilled water, in which the *H*-ion concentration is 0.0000001 or *N*/10,000,000, the *pH* value would therefore be the logarithm of 1/0.0000001 or of 10,000,000, which is 7.0. This value 7.0 is, therefore, the neutral point on the *pH* scale.

The *pH* values above 7 represent alkalinity and below 7, acidity. It does not follow, however, that a water with a *pH* above 7 will not be corrosive to both ferrous and non-ferrous metals. On the other hand, it does invariably follow that a water with a *pH* below 7 will be corrosive to both ferrous and non-ferrous metals. There have been instances where natural waters, having *pH* values as low as 4.5 were used regularly in air conditioning equipment but of course the spray washers of the air conditioning equipment required replacement after a few months operation.

Typical Analysis

The following represents the analysis of samples of water before and after use in washing air:

	Before Use	After Use
Free carbon dioxide (ppm)	6.0	54.8
Total solids (ppm)	40.0	217.0
Calcium sulphate (ppm)	13.0	132.0
(*pH* value)	6.9	4.8

It will be readily recognized that in the case of the water which has been used for washing air a distinctly acid water with highly corrosive properties has resulted. The analysis of this water shows that its characteristics are entirely different from those of the water originally entering the spray chamber in question. What brought about this change? The water in question had merely been used to wash several hundred thousand cubic feet of air and this air, aside from containing a certain amount of insoluble matter, was just *plain air*. A study of the foregoing analysis discloses the fact that the water now carries a high sulphate content. How did this originate? The burning of sulphurous fuels results in the formation of sulphur oxides which in combination with moisture form the corresponding acids. These acids were absorbed by the water and account for the high sulphate content.

Water used for washing air may absorb relatively large amounts of carbon dioxide forming carbonic acid, a corrosive agent, thus accounting for the high CO_2 content of water used for washing air.

It will be readily seen from the foregoing that all water used for washing air absorbs acidic matter from the air washed. Therefore, proper steps should be taken to see to it that the water used does not become actively corrosive through chemical conditioning of the water.

Heat Transfer Efficiencies

In the case of all waters, whether they be corrosive or scale forming or both, the problem of lowered heat transfer rates will be encountered. Condenser coils or Baudelot coils coated with rust or scale or combinations of rust and scale mean materially lower rates of heat transfer with a consequent increase in operating costs. In this connection it was recently shown that the amount of rust accumulating on steel coils over a period of a few weeks was sufficient to lower the heat transfer rate of the steel in question to that of glass. Since the heat transfer rate of glass is very low as compared to steel it will be apparent that the heat transfer efficiencies of metals coated with products of corrosion or with scale are also relatively low.

Another factor in this problem of water supply which is becoming of increasing importance is that of regulatory measures which are now being introduced by various municipalities for the conservation of the water supply. This means the use of recirculated water on the condenser systems of air-conditioning equipment.

The schedules maintained for discarding recirculated water vary widely. The schedule followed in the case of two separate air conditioning units recently inspected, which were located in the same building, will serve as an illustration. Both of these systems were operating on recirculated water for spray washers and for condensers. In one of the systems it was the practice to discard all recirculated water weekly and wash out any insoluble matter which might have collected during the operating period. In the case of the other system no regular schedule of discarding the recirculated water was carried out. At the time inspection was made, the water being used had been recirculated for a period of 41 days in one of the systems and for five days in the other system. The virgin water used had a total dissolved solids content of 60 ppm; after five days' use the total dissolved solids content had increased to 110 ppm; after 41 days' use the total dissolved solids content had increased to 285 ppm.

It is obvious that such radical changes in the analysis of the water used will bring about corresponding changes in the corrosive or scale-forming properties of the water. Actually it was found in the case of the system where no regular schedule for discarding water was used that the efficiencies of the system were lower than those obtained in the system where schedule of discarding water weekly and cleaning out system had been followed.

Location and Design of Air Intakes

Another interesting factor which is often of prime importance in this problem is the location and design of air intake ducts. It is natural for the architect to call for the location of such ducts at points which will conform to the architectural scheme developed. This will often result in locating ducts at points where excessively polluted air will be used. For example, ducts have been located at points close to smoke stacks and prevailing winds have at times blown stack gases into air intake ducts.

An unusual source of pollution was recently discovered. An air intake duct had been placed a few feet away from an exhaust duct leading from a storage battery room. The air from the storage battery room contained certain amounts of sulphuric acid mist which was sucked into the air intake duct leading to the air-conditioning equipment. Such a condition, unless immediately remedied, would lead to rapid deterioration of air conditioning equipment. In the case in question, remedial measures were taken to condition the water so as to neutralize the acid absorbed.

Conditioned Water for Correcting Corrosion and Scale Formation

In the preceding discussion the statement has been made that chemical conditioning of water would alleviate troubles caused by corrosion and by scale formation. The question may now be logically asked, How can water be chemically conditioned to alleviate corrosion and scale formation?

The corrosive action of certain municipal water supplies has been the cause of concern for many years. Ten years ago an effort was made to reduce the corrosive action of the water supply of Baltimore, Maryland, by chemical conditioning. The virgin water of this city is very soft having a low total

alkalinity and a *pH* value of 6.8. The water was treated with lime, thus increasing the total alkalinity and raising the *pH* to 8.5. It was found that such a treatment brought about the deposition of a thin film of calcium carbonate on piping through which the water flowed and that this film acted as a protective medium against the corrosive action of the water. This method of treatment is still in use in Baltimore and has been adopted by other municipalities. The treatment is open to this objection, it increases the soap hardness of the water. The reason for citing this treatment has been to set forth the possibilities of chemically conditioning a water so as to make it non-corrosive.

The chemist is not confined to the use of lime alone for such purposes. Proper control of the total alkalinity and *pH* of a water may be obtained by the use of various alkaline substances. In general, *pH* values should be maintained at levels between 8.0 and 9.0 and the alkalinity should be maintained at levels which will result in the formation of a protective film and at the same time prevent building up of a film to the point where heat transfer rates will be adversely affected. The problem must be handled so as to prevent either over-treatment or under-treatment of the water. The mere dumping of an alkaline material into the water will prove not only inadequate, but definitely unsatisfactory and may give rise to difficulties more serious than those it is called upon to remedy. On the other hand, a properly regulated treatment with regular analysis both of treated and untreated water will afford optimum protection both from the standpoint of scale formation and corrosion.

Importance of Water Analysis

The wide variety of characteristics exhibited by the various waters used in air conditioning equipment does not permit going into detail in connection with the problem of chemical conditioning. Each water adopted for use must be analyzed. In addition, the same water must be analyzed after use and changes in its composition noted. Such a study will make possible the prescribing of proper treatment for the particular conditions encountered. Generalities are always dangerous. This is especially true in the case of water treatment problems.

Users of air conditioning equipment are prone to criticize the manufacturer of the equipment for troubles which may arise and to hold the design of the equipment as being responsible. It is the writer's belief that, taking into consderation the advanced stage of development of air conditioning information, the well established and reputable manufacturers of air conditioning equipment are seldom responsible for difficulties encountered, such as rapid depreciation of equipment, caused by corrosion. If the operators of air conditioning equipment will give some attention to the matter of water analysis and water conditioning, many of the troubles now encountered will be solved.

It should be pointed out that corrosion problems are not confined to those parts of air conditioning equipment in which water is used. Corrosion takes place in air ducts, fans, eliminator plates and other parts of air conditioning equipment which are used primarily for transfer of air. For example, gases such as hydrogen sulphide, which are sometimes encountered in the rayon industry, have given rise to corrosion problems of air transfer equipment.

SUMMARY AND CONCLUSIONS

This paper discusses the problem of water supply as related to air conditioning equipment. It is pointed out that characteristics of waters used vary widely. Certain waters are corrosive, other waters are scale forming. Waters often bring about high rates of depreciation of apparatus and lower operating efficiencies. It is shown that chemical analysis of waters followed by chemical conditioning will obviate many of the difficulties encountered.

No. 950

AIR SUPPLY, DISTRIBUTION AND EXHAUST SYSTEMS

By S. R. Lewis * (*MEMBER*)
Chicago, Ill.

NO MATTER how efficient may be the preparation of the air exterior to a room, the heating, ventilating or air conditioning plant is likely to be condemned unless the air distribution within the room is effective. Arrangements which have proved satisfactory for heating, often fail when used for cooling. While for small and old rooms it is customary and expedient to install portable cooling units, independent of the built-in heating and ventilating apparatus, there is no reason why all newly-designed heating and ventilating apparatus should not be adapted equally well to cooling and ventilating.

For the purposes of this paper, rooms in which air is used to supply or to remove heat and moisture are classified roughly as follows:

1. Residence and other comparatively small rooms; indirect warm air and/or conditioning system.
2. Residence and other comparatively small rooms; principal heating accomplished by direct radiators or convectors.
3. School building classrooms.
4. Theaters.
5. Rooms of large area but low head room, such as general offices, manufacturing rooms, etc.

Residence and Other Small Rooms

Under this heading are treated comparatively small rooms where the heating is accomplished (1) by a warm air heating and/or conditioning system (Class 1), and (2) by direct radiators or convectors (Class 2).

Heating by Warm Air System (*Class* 1)

It has been the practice with gravity circulation, to place the supply registers in the floor or in the baseboard at the floor, and to place the return grille in the floor of the first story. When mechanical circulation was introduced, any change in register locations met with considerable resistance.

One radical innovation was to extend the supply ducts to the outside walls and then upward in these walls so as to discharge the air horizontally into the rooms at about knee height, under the center of each major-sized window.

* Consulting engineer.

Presented at the 39th Annual Meeting of the American Society of Heating and Ventilating Engineers, Cincinnati, Ohio, January, 1933, by A. V. Hutchinson, Secy.

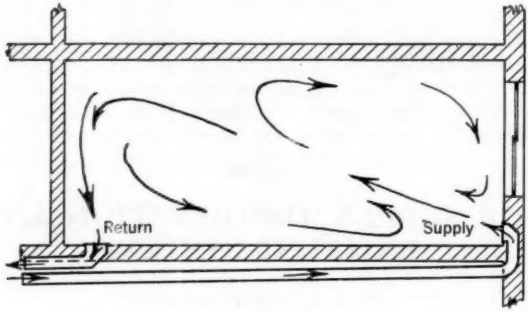


FIG. 1. AIR CIRCULATION WHEN HEATING WITH LOW-SUPPLY AND RETURN OPENINGS

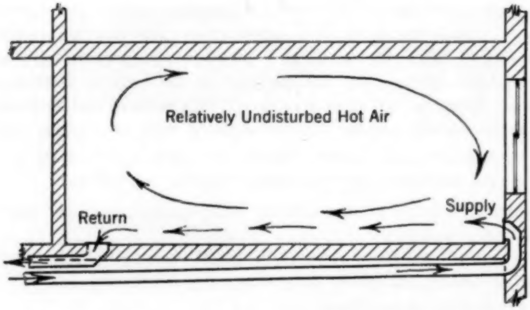


FIG. 2. AIR CIRCULATION WHEN COOLING WITH LOW-SUPPLY AND RETURN OPENINGS

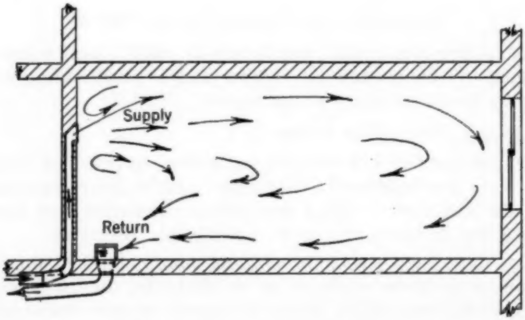


FIG. 3. AIR CIRCULATION WHEN COOLING WITH HIGH-SUPPLY OPENING AND LOW-RETURN OPENING

This was not entirely satisfactory as the occupants noticed the air currents. Furthermore, the riser in the outside wall was difficult properly to insulate and this location of the intake accentuated rather than alleviated down-drafts from the windows. The return air grilles with this arrangement of supply registers were placed at a central point, say in the hall, at or near the floor, or individual return grilles from each room were provided, usually at the side of the room opposite the supply register. While these arrangements operated more or less effectively for heating (Fig. 1), they were not satisfactory for cooling (Fig. 2). If cool air is introduced at one side of the room at the floor, and if the escape opening for the heated air to be displaced by the cool air is at the floor at the other side, the cool air will travel across the floor and will escape through the vent or return air opening, and will fail to affect the over-heated air in the upper part of the room to any perceptible extent.

Based on present available information, it appears that the air supply opening will serve satisfactorily if located high on an interior wall, looking toward the exposed wall, and that this location answers well also for gravity indirect heating. The return air arrangements corresponding to this apparently are not subject to exact rules, but must be adapted to circumstances. For instance, where the building is compact, with a first story having rooms open to each other, a single, centrally-located return at the floor functions satisfactorily for heating, and if the second story bedrooms are also compactly arranged no individual return from each will be necessary. On the other hand, any room which is unusually exposed or which is especially remote with reference to the other rooms should have a tight, controlled return grille and duct. This return preferably may be, with a mechanical system, close to the floor below the supply grille, and with a gravity system may be close to the floor at the opposite side of the room from the supply grille.

For a combined mechanical heating and cooling system using refrigeration for cooling, no particular change in the ducts usually is necessary. It is desirable from an economic standpoint to take advantage of the natural tendency of the cooler air to remain below the warmer air over head, and anything which will bring about such stratification will effect an economy in refrigeration.

When a mechanical warm air system has adequate return ducts, appreciable cooling can be accomplished with natural means, as follows: The fan outlet usually has a by-pass duct leading to a basement window or to a chimney provided for the purpose. The return duct has an alternative shaft opening into the highest part of the house. At night, in summer, the fan may be operated to exhaust the hot air from the top of the house by the return air duct just described and the fan will blow this heated air out of doors through the window, or preferably, of course, through the chimney. The cooler night air must then enter the house through the windows, and by its motion and temperature will extract the heat from the walls and furniture. When the hot day-period approaches, the fan should be stopped and the windows closed.

Fig. 3 shows the air circulation when cooling with a high supply opening and a low return opening. The air circulation, when heating, will be substantially the same as when cooling. Fig. 4 shows a section through an elemental mechanical warm air heating-cooling system. The attic plan is alternative. Summer night cooling may, of course, be accomplished by placing an exhaust fan in the attic.

Heating by Radiators and Convectors (*Class* 2)

Rooms heated by radiators and which have ventilation, with air supply and exhaust openings, are encountered in office buildings and hotels, and also in residences to a limited extent. The off-hand observation of the experienced engineer will be that no satisfactory distributing arrangement for the air in such places has been devised.

The entering air may be comfortable with conventional methods of side-wall inlets, when the entering air is warmer than the air already in the room, but the radiators, lights and people always will do the heating, so that the

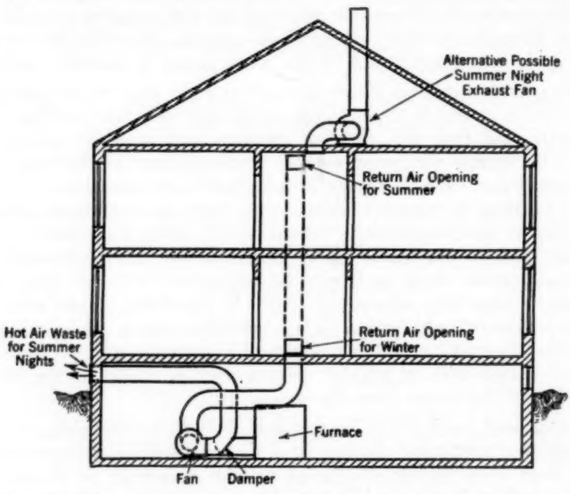


FIG. 4. SECTION THROUGH AN ELEMENTAL MECHANICAL WARM AIR HEATING-COOLING SYSTEM. THE ATTIC FAN IS ALTERNATIVE

ventilating plant practically always is a cooling plant and the entering air usually must be cooler than the air already in the room; with the introduction of this cool air comes draft-complaint.

The method of air introduction of Fig. 3 will serve for cooling in a residence, since the occupants may move if they feel the draft, but it fails in an office because the occupants may be desk-tied and cannot move. The draft condition with cool air introduced in a residence as per Fig. 3, exists only in summer when perhaps a draft would be a sought-for relief. If the cool draft from such an inlet carries on also in winter, as it surely does in radiator-heated rooms, the condition is unbearable.

Rooms in this classification are usually provided with glass chutes and elaborate diffusers, and generally they are closed up with make-shaft dams. When the air must be introduced, as is necessary in these rooms, at a temperature cooler for the entire year than that in the room, arrangements as per Fig. 5 have proved least objectionable. This air circuit has been found very

satisfactory in spaces up to 400 sq ft in area, with ceilings as low as 8 ft. No maximum ceiling limitation has been found.

The arrangement shown in Fig. 5 is adaptable for general or private offices, and for rooms having cold exposure with radiators, or for rooms with no windows or outside walls, heated by electric lights and people. It has been found satisfactory in very large areas used for general offices and for department stores where one inlet-outlet has been provided for each construction-bay center.

When the room in question is provided with a unit ventilator which obtains its air supply directly through the wall from out of doors, the problem of distribution is by no means easy, although with a high velocity air jet passing in an upward direction, much more tolerance may be expected than from an inlet like that shown in Fig. 3.

Experience seems to indicate that the use of unit air conditioners for summer cooling introduces no new features or difficulties which have not already been encountered in winter cooling. They must be provided with positive control by means of valves or dampers, or by both, which will prohibit any sudden and wide temperature variations, and by the time the entering air reaches the people, it must not be cooler than the air already in the space by a margin not yet definitely established, but in the order of 7 deg. It may be that this temperature-margin is dependent on the ceiling height of the room and on the velocity of the air at the discharge grille.

SCHOOL BUILDINGS

The air distribution conditions in school building class-rooms (Class 3) are not unlike those illustrated in Fig. 1 for mechanical warm air systems and those in Fig. 6 for unit ventilator-equipped plants. The thermostat (Fig. 6) which controls the mixing damper and the heating unit in the unit ventilator should be in the air stream from the machine. School rooms which have center-ceiling inlets along the lines of Fig. 5 have given excellent results. It is important that the temperature of the entering air, whether this air be supplied by a local unit ventilator or by a distant central fan, be controlled so that the air cannot enter the room from a side-wall inlet or from a unit ventilator, at a temperature more than a very few degrees cooler than that of the air already near the ceiling of the room.

Fig. 7 shows a section through a room equipped with a unit air conditioner or unit cooler. This is typical of the condition in effect when any recirculating room-cooling unit is installed.

In Fig. 8 the cloakroom ceiling is furred down so as to conceal the metal air supply duct, which is close to the ceiling. The temperature of the air for ventilation is controlled usually by a duct thermostat near the fan, at a temperature not more than 5 deg cooler than the temperature required by the room thermostat.

In Fig. 9 the direct radiator is controlled by a thermostat on the wall, and the air delivered by the unit ventilator is controlled by a second thermostat in the air stream from the machine. If the corridor is used as a vent flue, in accordance with questionable practice, no particular difference in the air circulation in the room will be observed from that of Fig. 9.

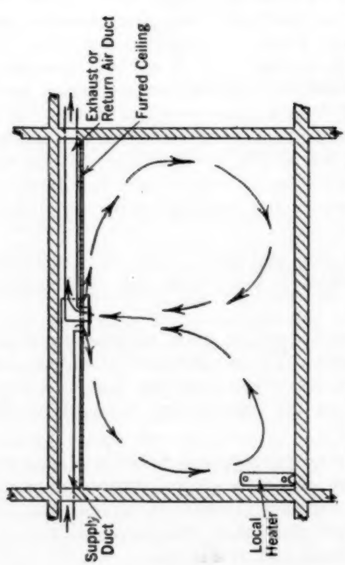


FIG. 5. SECTION THROUGH A RADIATOR-HEATED ROOM

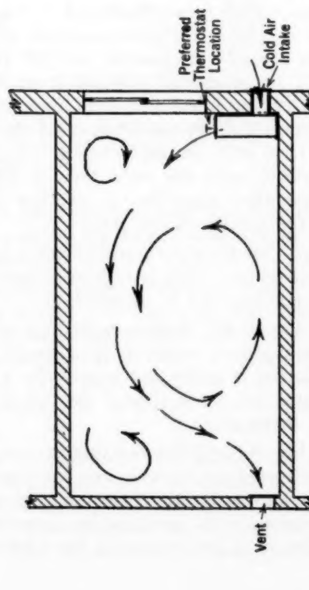


FIG. 6. SECTION THROUGH A UNIT VENTILATOR-EQUIPPED ROOM WHEN HEATING

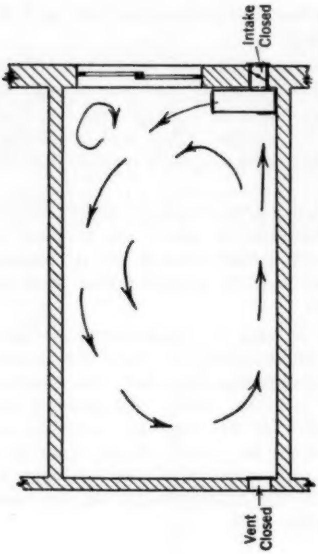


FIG. 7. SECTION THROUGH A UNIT CONDITIONER-EQUIPPED ROOM WHEN COOLING

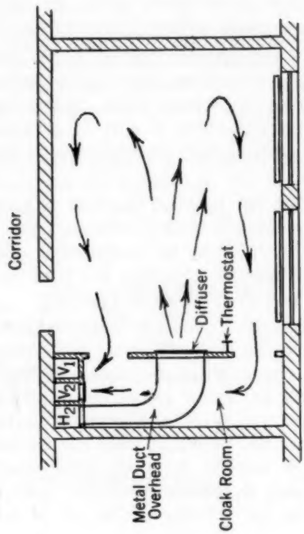


FIG. 8. PLAN OF A CLASSROOM IN A SCHOOL VENTILATED BY A CENTRAL FAN

THEATERS

Theaters (Class 4) are generally heated or cooled with air. No heating system for a theater should be given consideration without definite provision for cooling. Theater cooling always is far more important than theater heating.

There are two widely different kinds of theater air distribution, namely, the *upward* and the *downward*. The upward system probably is the older. When the entire seating area is occupied, the upward system gives little trouble when cooling, and as very little heating is required under such conditions, practically no difficulty is encountered. A maximum of about 25 cfm of air per person introduced at low velocity, say 150 linear feet per minute and a temperature not more than 6 deg below the room temperature, is indicated with an upward system.

If the auditorium is only partly occupied, the cool air, which is forced in under the vacant seats has nothing to heat it, and in theaters which have

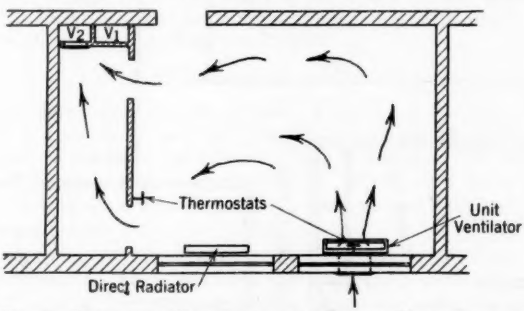


FIG. 9. PLAN OF A CLASSROOM IN A SCHOOL VENTILATED BY A UNIT VENTILATOR

sloping floors, this cold air falls down the incline and causes draft complaint from the patrons. This difficulty can be alleviated by an elaborate system of zone dampers and by painstaking manual control.

If no inlet openings are possible in the ceiling the upward system may be the least objectionable alternative. Fig. 10 shows a section through a theater with the upward system of air distribution. The patrons in the occupied zone often suffer from drafts due to the cool air which comes from the unoccupied zone.

Theaters under competent design usually are equipped with downward air distribution with horizontal diffusion of the entering cool air so as to combine it, both as to temperature and dilution, with the heated air which inevitably must rise from the bodies of the patrons. The waste or the recirculated air is withdrawn from the room at the floor. If the theater is large, and if the exhaust openings are placed in the side walls at the floor, drafts may be felt by the people who sit near the openings. There is no objection, however, except that of cost, to the use of small exhaust openings under each seat. These may have mushroom covers or may be cleanable floor grilles.

If the entering cool air in a downward system is not deflected horizontally, it will fall through the surrounding much hotter air, and will reach high velocities by the time it strikes the heads of the occupants. For instance, if air

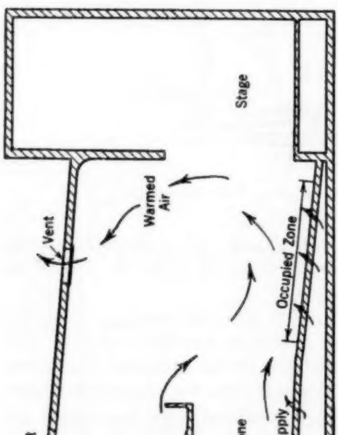


FIG. 10. SECTION OF THEATER WITH UPWARD VENTILATION

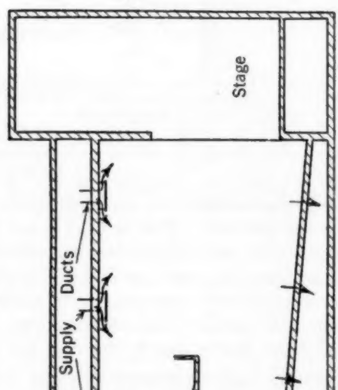


FIG. 11. SECTION OF THEATER WITH DOWNWARD VENTILATION

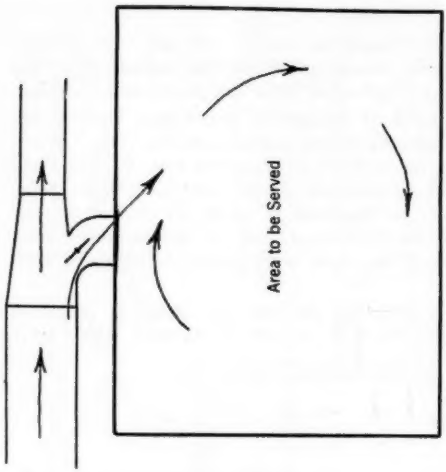


FIG. 12. PLAN OF CONVENTIONAL DUCT BRANCH

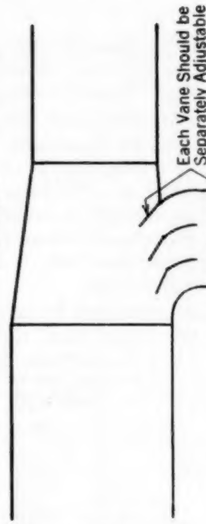


FIG. 13. PLAN OF IMPROVED DUCT BRANCH

at 65 F and 75 per cent relative humidity for cooling is released with practically no velocity, at the top of a theater in which the air has an average existing temperature of 75 F and a relative humidity of 60 per cent, the difference in weight per cubic foot between the existing air and the new air is 0.0015 lb.

If the ceiling at the air inlet location is 40 ft above the people, a column of air 35 ft high will be bearing down at the rate of 0.0525 lb per square foot more than the surrounding air. This weight is equivalent to 0.0099 in. of water and to a velocity of 380 fpm. Air forced over one's head at a temperature 10 deg below that of the surrounding air and at a velocity of nearly 400 fpm, is decidedly objectionable.

Fig. 11 shows a section through a theater with downward ventilation. There are deflectors which cause the entering cool air to be spread horizontally so that it will mix with the hotter air. The final escape is through well-distributed openings through the floor.

There have been cases in which the downward system of air distribution illustrated in Fig. 11, especially when not provided with refrigeration for cooling and not adequately controlled, gave trouble due to overheating at the rear, both above and below the balcony. It is especially necessary that adequate removal of the heated air be provided at these low-ceiling points and it is probable that auxiliary exhaust at or through the ceiling after the manner of the arrangements shown in Fig. 5 will be helpful.

LARGE AREAS WITH LOW HEAD ROOM

When large areas with low head room (Class 5) are to be ventilated, the inlets and outlets as shown in Fig. 5 may be used and, being overhead, have advantages due to freedom from risers which might interfere with partitions or which would increase the sizes of columns. When both the supply and the exhaust are distributed in this manner it is obvious that each construction bay is independent of every other similar bay.

In other cases it may be convenient to use overhead supply ducts with side-looking diffusers. These diffusers will not give a uniform and agreeable spread of the air for their dependent area unless carefully designed; it must be remembered that they receive air from the main line at comparatively high velocity, and that they are responsible for reducing this velocity quietly, and for distributing the reduced-speed air.

The conventional and objectionable branch outlet is indicated in Fig. 12. The butterfly volume damper, hinged at its center, always is objectionable. The high-speed air causes high velocity drafts and practically is uncontrollable over the dependent floor area.

Use of Vanes

Fig. 13 shows a plan of an improved duct branch. This has reference only to horizontal distribution. From the standpoint of comfort, the use of vanes behind grilles so as to cause the air to maintain an orderly and uniform travel, is important. Supply openings from vertical risers are elbows just as truly as though they had ducts at both sides, and vanes here are helpful.

Fig. 14 shows a section through a room air inlet at the top of a rising flue. Vanes can be used at this point advantageously. Fig. 15 shows a section through a room inlet opening at the top of a rising flue and indicates air conditions when no vanes are present.

Ejector Nozzles

In many cases it is advisable to take advantage of the tendency of any moving object to travel in a straight line, when designing air inlets. It is not

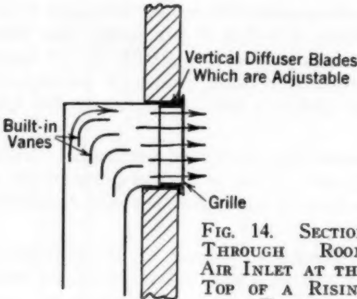
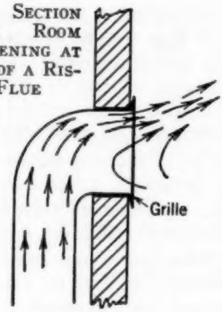


Fig. 14. Section Through Room Air Inlet at the Top of a Rising Flue

Fig. 15. Section Through Room Inlet Opening at the Top of a Rising Flue



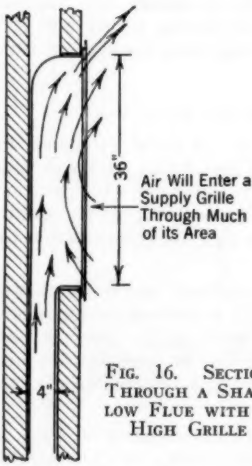


Fig. 16. Section Through a Shallow Flue with a High Grille

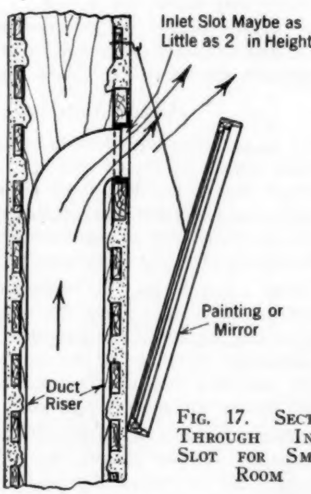


Fig. 17. Section Through Inlet Slot for Small Room

good practice to design a duct say 36 in. wide and 4 in. deep for a velocity of say 1,200 fpm, with a grille on that duct, say 36 in. x 36 in. The outlet velocity air emerging from a duct is taken advantage of, and ejector nozzles illustrate this fallacy.

In many theater and commercial installations this ejector-like effect of high-velocity air emerging from a duct is taken advantage of, and ejector nozzles scientifically proportioned are installed so as to cause definite recirculation of the room air due to the entering air.

Narrow Slot Inlets

In rooms of Class 1 in which it usually is expedient, if not always necessary, to use riser ducts 4 in. or less in depth the inlets to the rooms may be narrow slots which sometimes may be entirely invisible behind pictures and the like, as suggested in Fig. 17.

No. 951

COW BARN VENTILATION

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(*MEMBER*)
NEW YORK, N. Y.

THE principles, applications and details of cow barn ventilation differ from the ventilation of buildings used for human occupancy. A collection, digest and review of available information, both from technical and manufacturers' sources should therefore be of interest.

THE COW

All domesticated cattle in the United States are of European origin, of the *genus Bos, Bos taurus* being the dominant species. The historic background and climatic adaptability of the cow indicate that she is native of cold regions. When well fed and properly housed, cows enjoy and thrive in cold climates. For these reasons, the principal dairy areas of the world are confined to the colder sections, although dairying is gradually assuming more importance in California and the South.

The domesticated cow has existed since recorded time. The earliest written known records, including the Old Testament of the Bible, refer to cattle. The cow of today is therefore a creature of environment unnatural to her and of habits acquired through her long contact with man.

The size of the cow varies with the species, ranging from the smallest, the Kerry cattle of Ireland averaging 400 lb in weight, to the largest, the Great Shorthorns, whose show bulls and steers have attained weights as high as 2,500 lb. When full grown, the males are always the larger and weigh about 50 per cent more than the females of the same breed.

Breeds of cattle are usually classified into three groups, according to their functions and use, namely, (1) the specific beef group, raised for their meat production; (2) the special milk group, for their milk yield; and (3) the general or dual-purpose group, used for both meat and milk production.

Of the approximately 20 breeds of cattle found on American farms, only nine can be classed as of importance so far as general use and number are concerned. The four supreme breeds in the special beef group are the Shorthorn, Hereford, Aberdeen-Angus, and Galloway. In the special dairy group, the five important breeds, together with the average weights for full grown cows, are Holstein, 1,250 lb, Jersey, 850 lb, Guernsey, 1,050 lb, Ayrshire,

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1,100 lb, and Brown Swiss, 1,200 lb. These vary in the amount of milk and percentage of butter fat produced.

NEED OF VENTILATION

To one not entirely familiar with modern dairy management, equipment and care, the question might arise as to the necessity of artificial ventilation of cow barns. Modern dairy practice shows that at outside temperatures below 50 F, milk production is affected. While 50 F is not the minimum allowable barn temperature, it is generally customary to start to stable cows when the temperature during the night is likely to drop below that temperature. With outside temperatures above 32 F, barn windows and doors may be kept partly open, but at temperatures below freezing they must be closed and the ventilating system operated at full capacity. During extremely cold weather, the amount of ventilation may be somewhat curtailed to conserve heat in the barn.

The old-fashioned barn—with its ineffective, if not entirely lacking ventilating system, insufficient or improper window sizes and locations which retarded the entry of the health-giving sun, with damp and dripping walls, floors and ceilings, foul smelling and unsanitary—resulted not only in curtailed milk production, but encouraged pneumonia, pleurisy, and tubercular cows. Such barns, especially where commercial, have long been outlawed in the interests of public health. To remove the moist and foul air and disagreeable odors necessitates a properly-designed, controlled, and continuously-operated ventilating system.

The purpose of a barn ventilating system is not only the removal of moisture and odors, but also the supplying of the necessary amount of fresh air required for the needs of the cows. A producing cow needs an abundant supply of fresh air, especially when under the increased feeding necessary for greater output of milk or for increase in weight. She needs this fresh air to digest and assimilate her food and produce animal heat, to repair and build up body tissues, and to make a plentiful supply of milk.

To have and to retain healthy stock requires sanitary living quarters, proper and well-balanced rations of food and water, and an abundant supply of fresh air free of drafts. Vitality is the quality that makes milk records. Inadequate ventilation will reduce the vitality of cows and make them more liable to disease. Insufficient ventilation is also one of the principal causes of smelly barns with damp and dripping walls and ceilings.

AMOUNT OF VENTILATION

The problem of ventilating cow barn buildings differs from that of ventilating buildings used for human occupancy in that the spaces to be ventilated have higher relative humidities with comparatively low air temperatures.

Excepting so far as air volume and consequently building exposure area per head of cattle affect temperature, large cubical air content of a barn does not determine the purity of the air. Continual air changes and movement are necessary. As the loss of heat from the body of the cow by evaporation is affected by both the temperature and relative humidity of the surrounding air, the amount of ventilation not only affects the rate of removal of moisture, but also the rate at which moisture is produced.

The air in a barn is vitiated by the animal excrement, odors of certain feeds, and the products of respiration. Odors detected in the milk are not always caused by the feed, but may be absorbed from the contaminated air in the barn. A successful barn ventilating system is therefore one which continuously supplies the proper amount of air required by the stock, with proper distribution and without drafts, and removes the excessive heat, moisture and odors, thus maintaining the air at the proper temperature, relative humidity, and degree of cleanliness. As with humans, the physical comfort of stabled cattle depends upon the temperature, humidity and air motion.

The original air quantity standard proposed by the late Professor King[1] was based on the amount of pure air which had to be breathed by one cow, and on a standard of air purity recommended by him. His computations gave 117 cfh as the amount of air required by a cow averaging 1,000 lb in weight, and his recommended standard required a purity of the air in the barn of 96.7 per cent, that is, the air in the stable was not to contain more than 3.3 per cent of the air once breathed. This gave 3,542 cfh of air as the amount of ventilation required by one cow weighing 1,000 lb. As air expired from the lungs contains 4.24 per cent by volume of carbon dioxide, and pure air 0.028 per cent by volume, King's standard would allow $4.24 \times 0.033 + 0.028 \times 0.967$, or 0.167 per cent by volume of carbon dioxide. Armsby and Kriss[2] suggest that the amount of air to be supplied per cow is more correctly 3,452 cfh.

In determining these two standard ventilating air quantities based on CO_2 production, little consideration was given to the air temperatures and relative humidities. M. A. R. Kelley,[3] basing his calculations on air temperatures and relative humidities, states that 2,700 cfh of air per cow would be sufficient to remove the average production of 4,375 grains of moisture given off by a cow producing 20 lb of milk per day. His calculations are based on an outside temperature of 32 F and a relative humidity of 85 per cent and a barn temperature of 53 F and a relative humidity of 75 per cent.

Strahan [6] proposes the following values for volume of air flow in cubic feet per hour per cow. These values are also based on air temperature and humidity control, but in addition, consider the variation in minimum outside temperatures found in the different dairy zones:

Zone	Normal	Restricted	Theoretical Required to Prevent Condensation
1	2600	2000	2163
2	3175	2600	2225
3	3550	3000	2225

The normal amount of ventilation proposed is based on the daily mean temperature obtained in each zone. (See table of temperatures for January and February.) When the outside temperature falls below this daily mean, the amount of ventilation must be restricted so that the thermal balance is maintained.

Temperature of Barns

Nature provides such birds and animals which do not migrate to warmer climates on the approach of cold weather with more feathers or a more

abundant and longer hair growth as a protection against the cold. Therefore, when stabling animals which part of the time are outdoors, too much of a variation between the two temperatures should not occur. The vitality of cows is reduced when they are kept in barns that are too warm.

The comfort and condition of a cow affects her milk production. As an increase in milk production requires an increase in feeding, the barn temperature should be low to induce the cows to eat more, but not so low that food is wasted for production of unnecessary body warmth.

While a barn that is *too cold* reduces milk production and consequently results in a waste of feed, a barn that is *too warm* will cause a loss in appetite and a corresponding decrease in milk production. The amount of milk produced is further affected by sudden changes in temperature. A cow barn should be at a temperature high enough to prevent the freezing of the water in the drinking cups and provide comfort for the men working in the barn. Furthermore, the temperature should be high enough to permit the supplying of the required amount of fresh air during cold weather without producing discomfort to the animals.

Investigation shows that barn temperatures below freezing and above 80 F affect the milk production, decreasing both the quantity and quality of the milk. Animal comfort does not require high barn temperatures. Barns containing milk-producing stock should be kept at a temperature between 45 and 50 F. For dry stock, when at reduced feeding, the barn may be 5 to 10 deg higher. Calf barns are generally kept at 60 F, while hospital and maternity barns have a temperature of 60 F or somewhat higher.

ANIMAL HEAT

Food is the source of animal heat. The heat therefore given off by a living being depends on the amount of food consumed and the efficiency with which the heat generated is conserved. Armsby and Moulton[4] state that a dairy cow uses about 70 per cent of the combustible food energy absorbed. Knowing the amount of food consumed in a given time and its calorific value, the amount of heat produced by a cow can be estimated. The heat produced by the food must then be equal to that used for body gain (flesh, fat and milk), plus all the heat losses from the body.

The heat from the body is lost through respiration, by elimination, and by losses from the skin by evaporation, radiation and convection. Howell[5] quotes Vierordt as estimating the total heat losses from the body as follows:

Urine and Feces	1.8%
Expired Air	
Warming of Air	3.5%
Vaporization of Water from Lungs	7.2%
Evaporation from Skin	14.5%
Radiation and Convection from Skin	73.0%
	100.0%

From the foregoing figures it will be noted that with the exception of the first item, about 98 per cent of the total heat given off by the body is affected by the thermal conditions of the surrounding air. Hence the importance of temperature, humidity and air movement in producing body comfort.

The amount of food consumption does not only vary among the different

species, but also varies among animals of the same kind. As is noted in humans, the amount of food actually required varies with the age, sex, weight, temperament, physical conditions, and the physical and digestive activity of the individual. The thermal conditions of the surrounding air also has its effect on the appetite. Other things being equal, an increase in weight or physical activity should increase the amount of food consumed.

It has even been found that the posture of an animal has its effect on the amount of heat produced. For example, a standing animal, due to the greater muscular activity, gives off more heat than an animal lying down. Kelley[3] has found that with cows this increase in heat production is sufficient to raise the barn temperature, under average conditions, 1 to 2 deg in one-half hour, this increase in temperature being most noticeable in the morning after the cows stood up.

To estimate the heat production of various farm animals under normal condi-

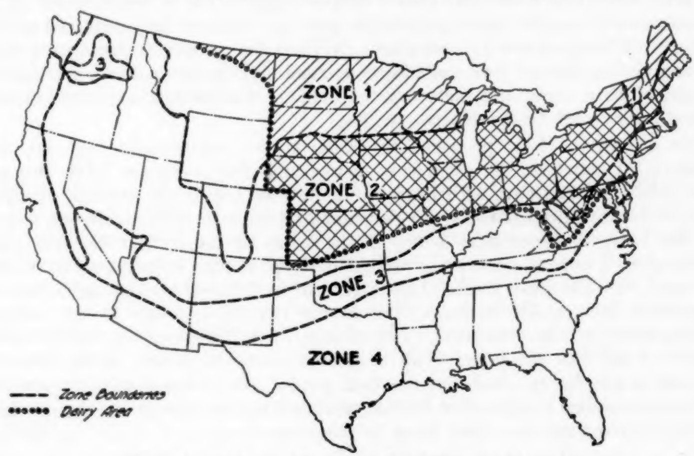

FIG. 1. CLIMATIC ZONES OF THE UNITED STATES

tions, Rameaux's Law may be used. This law states that in animals of the same species the calorification is proportional to the cutaneous (*i.e.,* skin) surface and to the cube root of the square of the weight of the body, together with the use of a suitable constant.

ALLOWABLE BARN CUBAGE

The design of a cow-barn building to be located in a cold climate so far as thermal conditions are concerned differs from a building designed for human occupancy in that for the former the heat available is fixed by the size and number of animals to be stabled, while in the latter the source of heat is variable and can be sized to properly take care of the given conditions.

The air temperature of a cow barn is the resultant of the heat generated by the stabled animals and the heat lost from the barn through the barn

structure, plus the heat used by the ventilating system. The cubical contents of a barn that can be heated and maintained at the desired air temperature by the stabled animals therefore depends on the animal heat produced and the heat losses just mentioned. Consideration must also be given to the actual space required for stabling the animals, including that required for the feed and litter alleys. The size of a stall depends on the breed of cow to be housed. For an average cow, a 3½ ft wide and 9 ft long (which allows for manger and gutter) stall should be provided. Feed alleys should be not less than 4½ ft wide, while litter alleys require a minimum width of 5 ft. In barns having two rows of cows facing in, the common feed alley should be not less than 6½ ft. Where the cows face out, the common litter alley is made 7 ft wide. Allowing for necessary feed and litter carriers, the height of the barn should be at least 8 ft. Accordingly, the minimum floor space required by a cow is 60 sq ft and the minimum cubage nearly 500 cu. ft.

Under conditions where the lowest outside temperature is never under 32 F, the volume of barn air space permissible per cow need not be considered so far as barn air temperature is concerned Where the outside temperatures drop below freezing, animal heat must be conserved and the question of permissible volume of barn space and therefore the area of the building exposure subject to heat losses must be considered.

The amount of heat produced by a dairy cow varies with her size and condition of lactation. A small cow in low milk flow averages 2,700 Btu per hour, while a large cow with heavy milk flow may give off as high as 3,700 Btu per hour. Approximately 75 per cent of this heat is available for warming the barn, the remainder being in the form of latent heat.(6) The heat produced by a cow of an average weight of 1,000 lb may be taken as 3,000 Btu per hour.(8) The heat produced by a cow of a different weight, according to Rameaux's law, varies in proportion to the two-third power of the weight, and therefore can be calculated. For example, a heifer weighing 500 lb would produce 1,820 Btu per hour. The weight of calves of course varies with the age, but it can be assumed that the heat production of three calves averaging six months in age is equivalent to that produced by one cow of average weight, while two yearlings are about equal to one cow.

Fig. 1 prepared by the United States Department of Agriculture, shows the climatic zones of the United States. It is based on the recorded average temperatures for the months of January and February over a period of 30 years at 100 selected stations. The principal dairy area is confined to Zones Nos. 1, 2 and part of No. 3, from and including the Dakotas east to the Atlantic coast. The daily mean temperature at 8:00 a. m. for the months of January and February and the average temperature for these months for the various zones are given in the table:(8)

	Zones			
For January-February	1	2	3	4
Daily Mean Temp (8:00 a.m.)	5 F	17 F	27 F	36 F
Monthly Mean Temp.	11 F	22 F	30 F	over 32 F

The average extreme is about 20 deg less than the monthly mean temperature. Due not only to varying outside temperatures, but also to differences in opinions and practice, the number of days the cows are stabled varies widely in different sections of the country. As cows should be stabled when the night

temperature is likely to drop below 50 F, this temperature may be taken as a basis for determining the average length of the stabling season.

From the equivalent number of head of stock, the total heat that will be produced within the barn can be figured. Knowing this and the average number of days the cows are to be stabled, the number of cubic feet of air space per cow can be calculated from the following formula:[3]

$$V = \frac{H \times k}{D} \tag{1}$$

where

H = the heat in Btu per hour given off by one cow equal in weight to the average of all the cows in the barn,
D = the average number of barn days per year,
k = a constant, which for well constructed barns is taken as 60,
V = the volume of air space, in cubic feet, that can be kept at a comfortable temperature by one cow and therefore allowable per head of stock.

For barns in locations having unusual outside temperatures, either lower or higher than the average, D in Formula 1 may be replaced by 300-5T, in which T is the mean temperature in degrees Fahrenheit for the month of January. The formula then becomes,

$$V = \frac{H \times k}{300 - 5T} \tag{2}$$

Based on assumed average numbers of barn days per year of 300, 250 and 240, and a mean of 3,000 Btu per hour emitted by a cow averaging 1,000 lb in weight, the permissible barn volume per cow for Zone No. 1 would be 600 cu ft; for Zone No. 2, 700 cu ft; and for Zone No. 3, 750 cu ft. These figures, theoretically, may therefore be used, providing the heat losses through the structure are not so high as to upset the heat balance. Barns having the permissible cubage but not fully stabled or having building heat transmission losses greater than permitted, require either additional insulation or even artificial heating.

Kelley [3] has demonstrated by test that in a well-built barn, fully stabled and with an air cubage equal to that allowed for the outside temperature, it is possible to obtain a temperature of 40 F even when the outside temperature is as low as −30 F.

HUMIDITY OF THE BARN AIR

The increase in the amount of moisture found in barns over that present in the incoming air, is caused by the stabled animals. In cows, the principal source of moisture is from respiration and not evaporation. The cow does not possess sweat glands to the same extent as do human beings, these glands being mostly in their nose. The total amount of moisture or absolute humidity in the barn air therefore depends on the number and size of the animals housed, on the humidity of the incoming air, and amount of ventilation obtained.

With a fixed amount of moisture produced in a barn that is well and properly constructed, theoretically the absolute humidity of the barn air should vary with the amount of ventilation provided. In actual practice this does not exactly hold, as unaccountable leakage will affect this relation, but to a certain extent the amount of moisture can be controlled by intelligent operation of the ventilating system.

With the amount of moisture constant, the relative humidity of course varies with the air temperature, decreasing in percentage as the air temperature rises. It is therefore possible to remove barn air moisture with outside air having a high relative humidity, even saturation, providing of course that the outside temperature is lower than that of the barn. Under average weather conditions, the mean relative humidity for the month of January is found to be 85 per cent.

Armsby & Kriss[2] give as the average rate of moisture produced by a cow giving 20 lb of milk per day, as 15 lb of water per day, or 4,375 grains per hour. To prevent an increase in the moisture contents of the barn air, 4,375 grains of water must therefore be removed per hour by means of the ventilating system.

To set a standard of permissible relative humidity for cow barns is difficult, as such a standard is affected by so many variable factors requiring consideration. Kelley[8] suggests for a stable temperature of 45 F an average relative humidity of 80 per cent as being satisfactory, with an 85 per cent relative humidity as a limit not to be exceeded. Tests made by him show that it is not difficult to obtain this suggested standard, providing other conditions are favorable. With a barn temperature of 53 F, a relative humidity of 75 per cent can and should be maintained.

Condensation: Drip and Protection Against Same

Damp and dripping barn ceilings and walls may be caused by insufficient and improper ventilation, poor and leaky barn construction, insufficient production of animal heat, too great loss of heat through barn structure or through lack of necessary building insulation. Air movement over the barn ceiling and wall surfaces affects the amount of condensation removed, the higher the air velocity, the greater the moisture removed and the less chance for condensation and drip to occur.

Cracks in otherwise well constructed barns, or in the insulation itself, will present points at which condensation and drip may occur. In ceilings constructed of wooden beams, forming an air space, care must be taken to prevent the circulation of cold air through same, otherwise condensation and drip is apt to occur on the ceiling below.

Obstruction to the free and unrestricted flow of the air caused by building projections, such as beams, presents the possibility of condensation and drip forming at these points. This is caused by the creating of pockets of non-moving air. Therefore, from a ventilating standpoint, the interiors of cow barns should be as free as possible of all building obstructions and projections.

On account of the high relative humidities and low air temperatures encountered in cow barn work, it is not always possible entirely to eliminate the forming of frost in ventilating ducts and flues. To reduce this to a minimum, both supply and exhaust ventilating ducts and flues located in outside walls or passing through cold spaces must be insulated and made air tight to prevent condensation and drip. Air leakage in gravity exhaust flues will of course also affect the efficiency of such flues, and reduce the amount of air handled. The lower parts of horizontal ducts and also the vertical ducts, where excessive condensation might occur, should be made water tight by soldering.

Types of Ventilating Systems

Agricultural authorities generally agree that artificial ventilation is necessary for cow barns, but they differ as how to apply the ventilating system. While they agree that the air is not to be heated and that it be supplied through or at the ceiling, there is a difference of opinion as to whether the air should be exhausted from the ceiling or near the floor. Among manufacturers of farm building ventilating equipment there is a distinct and clear division between the two types, one strongly recommending bottom exhaust, while another with equal sincerity recommends the use of top exhaust. Therefore cow-barn natural-draft ventilation may in general be divided into two types, namely (1) top supply—bottom exhaust, and (2) top supply—top exhaust.

The first type of natural draft ventilating system, based on the theories of Professor King and known as the *King System,* supplies the air through the ceiling, generally along the center line of the barn, and exhausts the air from near the floor along both outside walls of the barn. With this type of system, it is customary to provide occasional ceiling openings, called heat doors, which are opened in warmer weather to permit the escape of excessive heat from the barn.

The second type, known as the *Rutherford type* of ventilation, originally supplied the air along the outside walls near the floor and exhausted the air through the ceiling in two rows, each row being located about one quarter of the width of the barn from the outside walls. A modification of this type, known as the *Modified Rutherford System,* supplies the air in the same manner as the King System, namely through the ceiling along the center line of the barn, but the air is exhausted through the ceiling near and along both outside walls of the barn.

Top Versus Bottom Exhaust

A movement of air, and consequently ventilation, will be obtained whether bottom or top exhaust is used. There may be certain conditions or results desired, making the use of one or the other type more desirable.

The air does not pass direct from the intake to outlet but, due to the heat radiating surfaces presented by the bodies of the cows, convection currents are generated, causing a swirling and mixing of the fresh and barn air.

Under general conditions, the air temperature at the ceiling of a barn may vary up to 10 deg higher than the temperature at the floor. Therefore, when the air is removed from near the floor the mean barn temperature would be higher than when the air is exhausted at the ceiling. This is borne out by the results of a number of independent tests which show that with bottom exhaust, other things being equal, higher and more uniform barn air temperatures are obtained, with increase in ventilation and reduced relative humidity. With types of ventilating systems having ceiling exhaust only, it is necessary during extremely cold weather to reduce the amount of ventilation so as to conserve the animal heat and prevent the barn temperature from dropping below the comfort point. With the exhaust openings located near the floor, larger volumes of warm air are held in the barn, resulting in better temperature control and less likelihood of sudden change in barn temperature.

GRAVITY AND FAN EXHAUST

The air supply required for barn ventilation is introduced into the building by temperature differential and wind pressure. The majority of air exhaust systems are also of the gravity circulation type, with the occasional use of an electrically-driven fan.

The amount of air moved in a gravity system is affected by the temperature differential and by wind action. A ventilating system designed for a certain air movement, based on certain thermal conditions, will not handle the desired amount of air when the temperature difference varies from that assumed. A fan operated exhaust system will overcome this difficulty and permit the

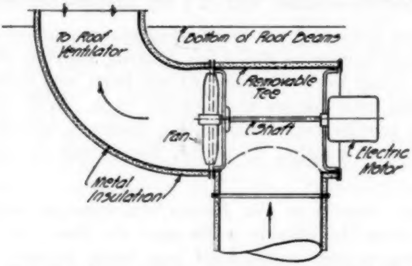

FIG. 2. ARRANGEMENT OF EXHAUSTER UNIT

moving of the desired amount of air, unaffected by temperature difference or wind.

The kind of exhaust fan generally used is of the disc or propeller type, with direct-connected electric motor. Electric motors should be totally enclosed as a protection against dust and moisture.

With either top or bottom exhaust discharging through roof ventilators, the exhauster unit is generally placed near the base of the ventilator. Fig. 2 illustrates such an arrangement.

Exhaust fans may be placed directly in the barn, exhausting either from the floor or ceiling and discharging through the barn wall. Hood and automatic louver protection should be provided on such air discharges.

Another type of exhauster unit, applicable to top exhaust, places the unit at the ceiling where usually the exhaust register or opening is located. Such an arrangement permits either gravity or mechanical operation of the exhaust ventilating system.

AIR INTAKES

The incoming air may be supplied to a barn either by means of windows or through wall and ceiling openings, the last two controlled either by hand or automatically.

Windows may be used as air intakes in the warmer zones of the country where freezing does not occur. They may also be used in colder climates as a supplement to the regular air intakes, when the outside temperature is higher than 32 F and an increase in the amount of air supply will not affect the barn air temperature so far as comfort is concerned. Their use in cold

climates is not desired as they will frost and drip, and they also require frequent adjustment and control for air temperatures and air quantities.

Air intakes located on the exterior of buildings should be provided with hoods, louvers, or other means permitting the air to enter freely, but excluding rain and snow. Furthermore, such openings should be protected by wire screens to prevent the entry of leaves, birds, rats, etc. Wire mesh smaller than ½ in. should not be used, as such screens clog up easily.

Under certain conditions, back drafting or reversing of the direction of the air flow will occur in the fresh air intakes. This may be caused by any of the following: by through draft, when intakes are located on opposite sides of the barn; when the air intake is on the leeward side of the barn, caused by change

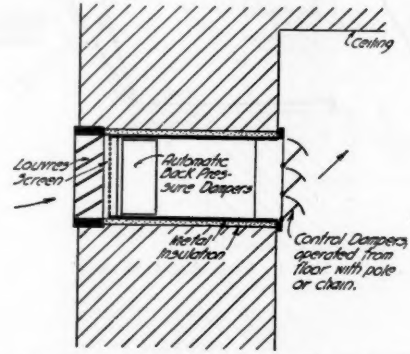

Fig. 3. Cold Air Intake and Wall Inlet

in wind direction and intensity; or when the air intake is located near the corner of the building or in locations where projections of adjoining buildings might deflect the wind direction. Kelley[8] states that back drafting will occur in wall intakes 6 feet or more in length at a wind velocity of 6 mph and with window intakes at 3 mph or even less.

Back drafting in air intakes may be partly overcome by lengthening the duct connection or by offsetting the duct in a vertical direction so that the exterior and interior openings are not opposite each other. Such arrangements will slightly decrease the air delivery due to the additional resistance introduced to the flow.

To completely overcome back drafting, back pressure dampers are used, which operate by the air current and automatically prevent the flow of the air out of the building. These dampers must be constructed of light and rust resisting material, such as aluminum, and should be free and easy swinging. They generally are placed in an angle frame and the edges of the dampers are provided with felt to reduce the striking sound and to prevent air leakage when the damper is closed. Fig. 5 shows such a damper swinging on a horizontal axis, while Fig. 6 indicates one having a vertical axis.

Figs. 3, 4, and 5 illustrate three different general types of wall air inlets and are self explanatory. It should be noted that each type is entirely insulated to

prevent condensation and drip in these connections. Fig. 6 shows a type where the air is introduced into the barn through the ceiling, the outside air being taken through any convenient opening located within a reasonable dis-

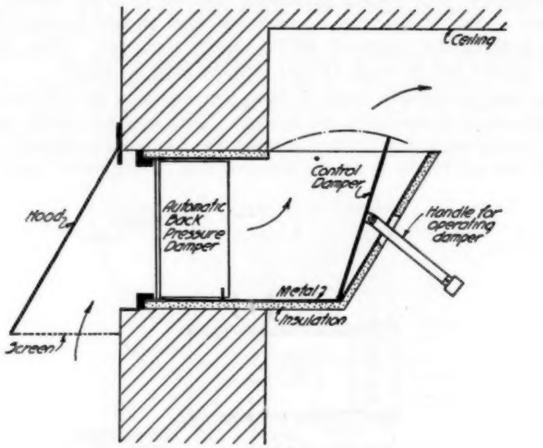

FIG. 4. COLD AIR INTAKE

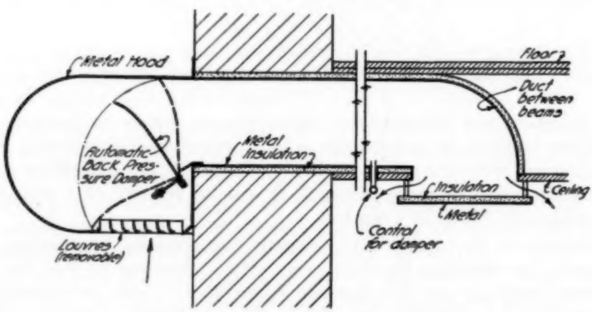

FIG. 5. COLD AIR INTAKE AND CEILING INLET

tance away, as a louvered dormer window. Several of these ceiling inlets may be combined and supplied from a common air intake.

AIR EXHAUSTS

The exhausted air is generally collected and discharged at the ridge of the roof through suitable ventilators. These roof ventilators may be of wood, having louvered and screened openings on all sides. If of metal, they generally are of the circular type. Fig. 7 shows a typical arrangement of such a ventilator. Ventilators of the revolving type are sometimes used to accelerate

the air movement by means of wind action. Such parts of the ventilator which are located within the building should be insulated.

Where the space over the barn is used for the storing of hay, vent openings should be placed at the base of the ventilator, to exhaust the air from the hay mow during the warm weather or when the hay is fresh and has just been stored and gases are likely to be formed. A movement of air will also prevent rotting of the hay should it become wet. Such hay-loft vent openings will reduce the amount of air exhausted from the barn proper. They should be kept closed in extremely cold weather as they are not needed or desired at that time. The mixing of the warm barn air with the comparatively cold air

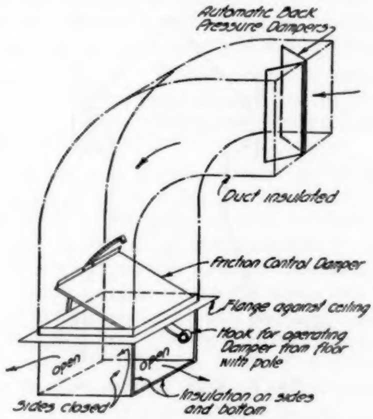

Fig. 6. Cold Air Ceiling Inlet

from the hay loft will cause condensation, with the possibility of the roof rafters rotting around the ventilators.

The use of hay and feed chutes for exhausting the barn air is undesirable as such an arrangement will upset the operation and control of the ventilating system. If air exhaust out-takes are properly located and arranged, no back drafting will occur and therefore no back pressure dampers are required in such connections.

Kind of Materials Used

In cold sections, the use of wood, even when insulated, is not recommended as such flues are not entirely free of condensation. The alternate wetting and drying of the wood would eventually rot it.

Flues other than wood should be of rust-resisting metals or some other kind of waterproof material. For metal, heavy galvanized iron, copper bearing galvanized iron, or copper are used.

All parts of the ventilating system exposed to the weather or where condensation may form, such as ventilators, wire screens, dampers, etc., should also be of heavy and rust resisting material.

HAND AND AUTOMATIC CONTROL

Each air supply and each air exhaust opening should be provided with friction dampers which can be placed at any degree of opening and be conveniently operated from the floor of the barn.

Automatic control systems have been developed and used in cow barn ventilation work. While such devices will control a system for the results for which the automatic regulators are set the setting and adjustment of them depend on the attention, judgment and experience of the attendant.

The damper on the exhaust outlet of a gravity system may be automatically controlled by a self-contained regulator, such as a bellows and weighted lever,

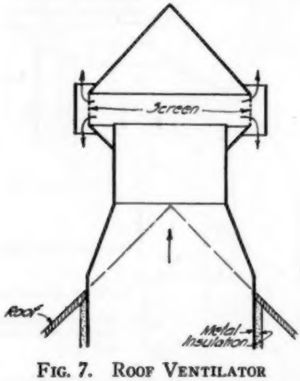

FIG. 7. ROOF VENTILATOR

adjustable for varying air temperatures. Dampers may also be controlled by electric or pneumatically-operated automatic controls actuated by thermostats located either in the barn proper or in the exhaust ducts. Where mechanical exhaust is used, the automatic control system would operate the exhaust fan motors.

BARN CONSTRUCTION AND INSULATION

The proper ventilation of the barn in cold weather is of major importance. Hence the heat losses through the structure proper should be reduced to a minimum so that as much as possible of the animal heat available may be used for tempering the incoming air, assuring maximum ventilation.

Barns should be of tight construction to prevent excessive leakage of air. Doors should always be of double thickness, the plys running across each other to prevent warping and buckling. In extremely cold climates, vestibules or storm doors are desirable. In such locations, storm sashes should be used on all windows to prevent excessive heat losses and the forming of condensation and drip. Single glass windows therefore should not be used. Due to low resistance to the flow of heat, single glass will frost easily, retarding the sanitary and heating effects of the entering sunlight.

Even in well-constructed barns, in which the heat transmission losses are within the allowable limit, care must be taken in building design and con-

struction to prevent the introduction of structural parts which will present low heat resistance. For example, along a concealed steel beam, which material is a good conductor of heat, the flange might be too close to the wall or ceiling surface, permitting condensation and drip to form at those points. In such cases, insulation must be provided to those parts. Silo, hay chute and other utility openings must be fitted with doors which can be closed air tight.

The necessity of insulation depends on a number of factors, such as outside temperature, the efficiency of the barn construction against the loss of heat, the lowest barn temperature, highest humidity, and the least amount of ventilation desired. Insulation is also indicated where the air cubage of the barn is greater than can be comfortably heated by the housed animals, or in barns where, although the air cubage is within the permissible limit, it is known or expected that at certain times the barn will not be fully stabled.

While it would be possible, to a limited extent, to increase the air temperature of a barn not properly constructed or insulated, by increased feeding, such procedure would not be economical in cost. The application of the required insulation is the better investment.

As the ceiling is not only at a higher temperature but generally has a lower heat resistance than the wall construction, special consideration as to the necessity of insulation should be given to this part of the barn. Where a ceiling is provided, thus covering the beams, the ceiling should be insulated as the space between the beams will get cold enough to cause condensation on an uninsulated ceiling. When the beams are exposed and the space over the barn is used for hay storage, the barn ceiling would be well protected against heat losses and no insulation would be necessary. But care must be taken in cold weather that all the hay is not removed from the hay loft floor, leaving it bare, as condensation may form on that part of the barn ceiling located below.

The efficiency and structural strength of the insulation to be used is of great importance. Consideration must also be given as to the effect of moisture on the insulating material, that it is fireproof, and that it will not harbor vermin, etc.

Permissible Building Heat Losses

The permissible heat loss through the barn structure depends upon the outside and inside temperatures, the available animal heat, the amount of ventilation desired, and the area of building exposure. For Zone No. 1, based on one cow, assuming 2,600 cfh of air for ventilation, 45 F barn temperature, 5 F outside temperature, and an available heat of 3,000 Btu (1,000 lb cow), the heat required for ventilation would be $2{,}600 \times 0.019 \times (45 - 5) = 1{,}976$ Btu. To maintain the desired barn temperature and amount of ventilation, the maximum amount of heat that can be lost through the barn structure is then $3{,}000 - 1{,}976 = 1{,}024$ Btu per hour for a 40 deg difference, or 25.6 Btu for a 1 deg difference. A typical barn having 600 cu ft of space per cow would have about 130 sq ft of building exposure permitting heat losses, such as walls, windows, doors, ceiling and part of floor. On this basis, the permissible heat loss per hour per cow, for a 1 deg difference, would be 0.043 Btu per cubic foot, or 0.197 Btu per square foot of exposure.

For Zone No. 2, based on 3,175 cu ft of air, a 50-F barn temperature, and a 17-F outside temperature, the permissible heat loss for a 1 deg difference

would be 0.051 Btu per cubic foot and 0.235 per square foot. For Zone No. 3 3,550 cu ft of air, 55-F barn and 27-F outside temperatures, the permissible heat losses would be 0.066 Btu per cubic foot and 0.305 Btu per square foot.

These permissible heat losses per square foot mean that the average heat transmission for all exposed parts of the structure, such as walls, windows, etc., must not exceed the figures given, if the desired barn temperature and amount of ventilation is to be maintained. At such times when the outside temperatures fall below the daily mean assumed, the amount of ventilation must be restricted to maintain the thermal balance.

SUMMARY

The following table is a summary of the data required in the design of a cow barn ventilating system, based on a unit of one cow, 3,000 Btu per hour heat available, 600 cu ft of barn air space, and 130 sq ft of building exposure:

Zone	Temperature Degrees Fahrenheit		Per Cow per Hour		
			Ventilation Cu Ft	Permissible Heat Losses per 1 Degree Difference	
	Daily Mean Jan., Feb.	Barn		Per Cu Ft	Per Sq Ft of Exposure
1	5	45	2600	0.043	0.197
2	17	50	3175	0.051	0.235
3	27	55	3550	0.066	0.305

For barns having a greater volume of space per cow than the 600 cu ft assumed, the permissible heat losses per 1 deg difference per square foot of building exposure must be reduced, while for barns with smaller volumes per cow this loss may be higher. In either case, however, the heat loss per hour per 1 deg difference per cow, must not exceed approximately 26 Btu (130 sq ft × 0.197) for Zone No. 1, 30 Btu for Zone No. 2, and 40 Btu for Zone No. 3.

ACKNOWLEDGMENT

The author wishes to express appreciation to M. A. R. Kelley, Agricultural Engineer, United States Department of Agriculture, and to W. B. Clarkson, Chairman of the Committee on Farm Building Ventilation, *American Society of Agricultural Engineers,* for their constructive suggestions.

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DISCUSSION

W. B. CLARKSON (WRITTEN): In attempting to complement the author's subject matter, it is not out of place to emphasize a few of the points of major importance that he has covered.

First, there used to be an opinion, quite general, that carbon dioxide could accumulate in a stock room to a poisonous extent, an erroneous theory long since exploded. As King states in his book, pages 8 to 24 inclusive, the things to emphasize are: air once breathed has lost much of its sustaining power; a continuous flow of air is necessary; and fresh air supply is certain to be inadequate at times if definite provision for it is not made. These are so important that too much emphasis cannot be placed upon them. On page 41 it is stated that the real problem involved is how nearly can the air of dwellings and stables be maintained at the normal out-of-doors fresh air purity with practical economy.

If ever there is a place where quality of air counts, it is in the planning and installation of a cow barn ventilating system. It is impossible to make the air of the stock room too pure. By this same token it is impossible to have the air of the stock room too dry for good health.

Second, another item of importance is the adequate insulation of the stock room walls and ceiling if one expects only to use the heat generated by the stock. This heating system has no opportunity to be expanded; the heat volume generated is held rigidly within narrow limits from day to day; it cannot fluctuate very much. Therefore, the building must be made to accommodate the heating plant. Every unit of heat must be conserved and used to maintain a proper heat balance as well as to use heat in the operation of the ventilating system.

Third, there is a natural difference in temperature rise of ½ to 1 deg from floor to ceiling in the stock room; hence placing flues to draw the air from the floor conserves heat in cold weather.

Fourth, the importance of maintaining absolute control of the air movement in a stable is of major consideration. Foul air should only pass through the flues designed for that purpose and should not be permitted to pass through the walls or ceiling at any other point. Fresh air should enter through intakes designed for that purpose and not be permitted to enter elsewhere. This, of course, applies to winter service when it is necessary to have heat control.

Since the floor is the coldest part of the room, a much larger volume of air can be removed per unit of heat lost by air moving out through the flues. Fresh air introduced at the ceiling avoids drafts on the stock, because the air diffuses into the warmer air at the ceiling, where it quickly loses its extra weight by being heated to the mean temperature of the room.

The general circulation of the room air is downward from ceiling to floor, where it carries off the foul air generated by the stock through the floor flues.

J. L. STRAHAN[1] (WRITTEN): The references cited by the author will be evidence to this group of engineers that the subject of stable ventilation has commanded the attention of agricultural engineers for a number of years past. To us who have worked on the problem, the names of King, Clarkson, Kelley, Fairbanks and others are significant ones, recognizing, as we do, the contributions each has made to progress in this field.

Now it is with much gratification that we mark the interest being shown by this Society, as evidenced by the author's instructive paper. That a practitioner of his standing has thought well enough of the problem to have worked up so admirable a presentation is an indication that we may henceforth look for a distinctly engineering and rational approach to a practice which has heretofore, to a large extent, though not entirely, been handled empyrically, by rule of thumb methods. Personally, I think this is a very decided step ahead.

The author has covered the subject in a comprehensive manner, and has shown how the available material may be utilized practically in design. To me the value of a discussion of this sort lies in the degree to which its findings may be applied in practice. So I would like to take this opportunity to amplify an idea that the author has touched upon in his final statements concerning design.

Engineering is largely concerned with the manipulations of inert materials. But into this problem there enters a rather unusual factor, a biological factor, living material. Because it is alive, instead of inert, makes it none the less an integral element in the solution. In fact, the proper coordination of building design, building construction, and ventilation design factors is based upon an expression of their values in terms of animal units. For instance: cows produce heat, and the heat is dissipated. All that is produced is lost in one way or another. A statement of this heat balance can be expressed as follows:

$$H = \frac{VD}{53} + ACD$$

In this formula, H equals heat available per animal unit per hour in Btu. $VD/53$ equals heat lost per hour through air change. ACD equals heat loss per hour through conduction and radiation. Both of the latter values are in terms of one animal unit.

It will be observed that V, or air change per hour, A, or area of building exposure per animal, and C, or rate of transmission per square foot per hour, are all coordinated here with the animal, as expressed by H, and climate as expressed by D or temperature difference. In other words, the building design (AC) and ventilation requirements (V) are all properly related to the animals to be housed, and the climatic conditions to be met.

We have found in our practice that adequate ventilation of a stable can be obtained only when its design and construction are thus properly adjusted to animal population and climatic conditions. And further, we have found that a statement of this adjustment can be made in terms of AC. This value merely states the rate at which heat is lost through the building structure by radiation and conduction per hour per degree temperature difference per cow. But in reality, it also expresses the design and construction of the stable in terms of its ability to conserve heat. It is, in effect, an index of the possibility of temperature control in the stable, when natural animal heat alone is available for use.

The *American Society of Agricultural Engineers* is working on the problem of standardizing this index, and proposals have been made as follows:

[1] Consulting Agricultural Engr., White Plains, N. Y.

Zone	Index (AC)
1	26.00
2	29.25
3	38.00 (approximate)

May I express my appreciation of this opportunity to discuss this paper, and also the hope that, as a result of the author's work, other papers will be presented before the Society giving additional new information as developed, thus keeping the subject up to date for the benefit of the profession at large.

A. M. Goodman [2] (Written): I am impressed with this paper because it shows plainly that the author has carefully studied every available source of literature on the subject of cow barn ventilation and has given unbiased judgment to various questions which, unfortunately, are debated by those writing on the subject. His bringing together of all available engineering data on the subject is a service valuable to every one having to do with the ventilation of shelters for domestic cattle.

There are, however, a few points which I feel have not been fully brought out. My discussion of these points is based on observations made by Prof. F. L. Fairbanks and the writer in our studies of the ventilation of dairy stables, which have been carried on in stables, over a period of 8 years.

First, we do not find the so-called back-drafting of inlet flues to be of any serious consequence, as this does not increase the amount of air going through the stable. At the time an intake is back-drafting, it is not acting as an intake flue, and while the air may be coming in rapidly at one side of the barn, this is compensated for by the fact that no air is coming in on the side where back-drafting is taking place.

We find also that it is essential that the intakes be placed along the side walls regardless of which way the cows face and also that the incoming air be directed straight upward. The reason for placing the intake flues along the side wall is that the cold incoming air then tends to join with and stimulate the convection currents within the stable.

The reason for directing the incoming air upward is, first, so that it cannot blow on the cattle, and second, so that it will immediately be widely diffused and mixed with the warmest air in the stable, that is, the air at the ceiling.

In regard to the exhaust flues, in barns up to 110 ft long, one out-take chimney is sufficient. This may be placed anywhere within the stable, provided that it extends directly to a point above the highest point of the barn roof. Since convection currents keep the air in the stable constantly in motion, it is evident that air from various parts of the stable will arrive at the exhaust point frequently. Studies of the air condition in barns housing up to 50 or 55 head of cattle, and using one out-take point, prove this contention to be correct.

Well built chimneys, made of two thicknesses of matched lumber with paper between, give a positive draft and condensation does not occur, provided, of course, that horizontal sections, sharp angles and restricted areas are avoided, and that a well designed wooden ventilator head is provided.

In regard to control of the ventilating system, we find that with out-take flues exhausting air from near the floor, there is practically never any occasion to throttle the ventilating system, though it be sufficiently copious in design to take care of the needs of the stock even in mild winter weather. If any throttling of the system seems necessary, we recommend that intake openings be throttled to a small extent, though never completely. The out-take flue, if exhausting from near the floor, removes only the coolest air in the stable, because, as the author has clearly stated, this air is often several degrees cooler than the air near the ceiling of the stable.

[2] Extension Professor, Dept. of Agricultural Engrg., N. Y. State College of Agriculture, Cornell University.

The air near the floor has practically the same composition as that in other parts of the stable.

K. W. STOUDER[3] (WRITTEN): Let me say as a veterinarian, that we could do vastly more in the control of many infectious diseases and improve general health of all livestock to a considerable extent, if the importance of proper ventilation was understood by the average livestock breeder and feeder. It has been my experience that the average breeder frequently has disaster in his herd either because of entire or almost entire lack of ventilation, or because in some instances he feels there is no ventilation in the building unless there is a very noticeable draft going through it. Of course this lowers vitality and consumes an enormous amount of energy from the animals living in it, so that either milk production or rate of gain is seriously retarded.

As an illustration of the damage done to livestock in a building inadequately ventilated, our experience in the control of tuberculosis in the State of Iowa for the past 10 years furnishes an excellent example. In the southern part of our state it has been customary to house animals in stables that are rarely completely closed and even in mid-winter there are few days when the animals cannot be permitted some outdoor living. Actual testing of all of the breeding cattle in these counties showed that less than 2 per cent of them were affected with tuberculosis and in many counties the per cent ran much lower than this. In the northeastern part of the state where the lay of the land permitted the frequent building of stables with at least part of the first story beneath the level of the ground, and where climatic conditions necessitated the housing of the animals day and night for sometimes a week continuously, and where practically no attention was ever given to the building of the stables first constructed there, tuberculosis became implanted in the herds. Through close contact in poorly ventilated stables, the disease distributed itself until we find frequently some of these herds infected to the extent of from 35 to even 50 per cent with tuberculosis. Slowly, livestock managers are beginning to better understand this problem and I believe, if economic conditions would permit them to purchase anything above bare necessities, an enormous number of them would be glad to install adequate ventilation in their buildings. This would not only reduce the danger of the spread of tuberculosis, but would make it much easier to rear young stock in these same buildings with less loss from pneumonia a common condition among calves where they live in close, damp stables part of the night and then are permitted to go out in the colder outer air.

Some years ago when a very high percentage of our swine were produced and fattened in open sheds, diseases of the influenza type were almost unheard of. During the past 15 years there has been an enormous increase in the construction of much better buildings for hog housing. Masonry walls with cement floors and tight roofs are frequently met with, and, on these farms, we often see the heaviest losses of so-called swine flu, because the animals are either closely housed during the night in this building and then compelled to go out in the cold open lots to feed, or because the buildings constantly provide strong floor drafts where the owner inadequately understands what proper ventilation of the building should consist of. This causes the animals to get inadequate rest through the night with the result of lowered vitality and therefore they are more susceptible to infection all the time.

[3] Extension Veterinarian, Iowa State College of Agriculture.

No. 952

AIR INFILTRATION THROUGH STEEL FRAMED WINDOWS

By D. O. Rusk,[1] V. H. Cherry [1] and L. Boelter [2] (*NON-MEMBERS*)
Berkeley, Cal.

THE rate of air infiltration through a window depends to a great extent upon the quality of construction of the window. For this reason, only the approximate magnitude of the leakage can be estimated from test data on similar windows. Experiments conducted at the University of Wisconsin [3] in cooperation with the A. S. H. V. E. Research Laboratory indicate a variation of leakage of from 50 to 90 cu ft per hour per foot of sash perimeter at a pressure equivalent to 25 mph wind velocity, for nine double-hung wood sash windows of similar workmanship. The width of crack perpendicular to the direction of air flow, the window size, and thickness were the same for these windows.

The A. S. H. V. E. Research Laboratory tested steel double-hung windows actually installed in a building.[4] The results of the University of Wisconsin experiments and those of the A. S. H. V. E. Research Laboratory, supplemented by further tests on swing and hung steel windows made at the University of California by the authors, are shown in Fig. 3. The leakage curves are designated by letters indicating the corresponding cross section through the leakage path. The infiltration rate for a window is specified as the volume of air leakage per minute per linear foot of crack for a given wind velocity normal to the window. The curves show the leakage as a function of the wind velocity. It may be seen that there exists a very wide variation in the performance of these windows.

Certain other variables not controllable by the manufacturer must be considered in performing tests on windows. The results given in the aforementioned papers [3-4] indicate a variation in leakage rates for latched and unlatched windows. It is an object of this paper to show that the test results obtained on steel swing windows depend upon the manner in which the window is closed and latched. Further, it has been noted that the leakage through felt

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[1] Assistant in Mechanical Engineering, College of Engineering, University of California.
[2] Associate Professor in Mechanical Engineering, College of Engineering, University of California.
[3] Air Infiltration Through Double-Hung Wood Windows, by G. L. Larson, D. W. Nelson and R. W. Kubasta, A. S. H. V. E. Transactions, Vol. 37, 1931, p. 571.
[4] Air Leakage Studies on Metal Windows in a Modern Office Building, by F. C. Houghten and M. E. O'Connell, A. S. H. V. E. Transactions, Vol. 34, 1928, p. 321.

Presented at the 39th Annual Meeting of the American Society of Heating and Ventilating Engineers, Cincinnati, Ohio, January, 1933, by P. D. Close.

weatherstripped steel swing windows depends upon the humidity of the air. Test results will be presented which indicate that for the window tested, the leakage was at least 130 per cent greater for dry air than for saturated air.

APPARATUS

The apparatus used for the tests, which were conducted in the Mechanical Laboratories of the University of California, is shown in Figs. 1 and 2. The

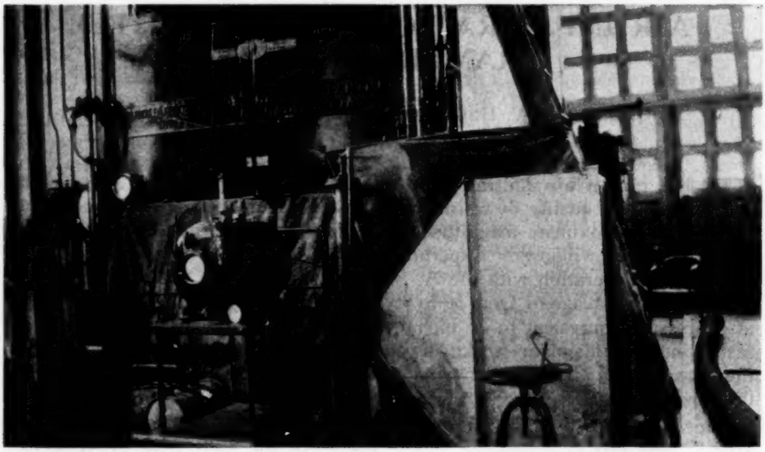


FIG. 1. ARRANGEMENT OF APPARATUS FOR INFILTRATION TESTS

windows and casings were mounted in a steel hood with the weather side of the windows facing the air inlet, which was located at the back of the hood.

The equivalent wind velocity in miles per hour was computed from the expression

$$V = K\sqrt{\frac{2gh}{w}} = K'\sqrt{\frac{h}{w}}$$

where

$K =$ a conversion factor $= 1.55$ or $K' = 12.45$.
$w =$ density of the air chosen as standard.
$h =$ the pressure head in the hood acting upon the window in inches of water. The head was obtained by means of an inclined manometer attached to the hood.

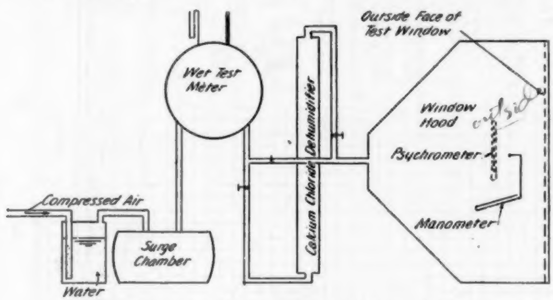
The equivalent velocity was calculated as the velocity of dry air normal to the window, at a pressure of 30 in. of mercury and at 60 F, corresponding to the actual pressure head acting across the window. Similarly the infiltration rate was calculated from the actual weight of air (and water vapor) flowing per unit time per linear foot of leakage space, as the volume of dry air at 30 in. of mercury and 60 F.

The air was supplied from a tank charged by a compressor. A wet test

meter was used to measure air quantity for the lower flow rates. This necessitated saturation of the air to prevent loss of water from the meter. A Venturi tube was used for the higher flow rates.

The piping from the meter to the hood was so arranged that the air could be passed directly, or through a calcium chloride container, to the hood. The humidity of the air was considered as a variable for the felt weatherstripping tests only.

Four steel swing windows and one steel double-hung window with brass weatherstripping were tested. One of the swing windows was felt weatherstripped. The felt seal was effected by cementing a strip of felt into a groove



inside

FIG. 2. DIAGRAM OF TEST APPARATUS

in such a manner that the window, when closed, was pushed tightly against the felt. The steel frame and casing formed the seal for the windows tested without weatherstripping. Figs. 3 and 4 show the cross section of the leakage path and the general location of the latches.

RESULTS OBTAINED

The effect of varying the method of closing and latching the door is shown in Fig. 5 for a non-weatherstripped window. Since the equivalent wind velocity is often used in specifying the infiltration rate of a window, the leakage rate (Q) is plotted against wind velocity for two conditions:

1. Window closed by action of lock only.
2. Window pulled tightly closed and locked.

It may be seen that, as the pressure on the window is increased, the window is closed tighter and remains in position as the pressure is reduced. The leakage for the first case is 10 per cent to 30 per cent greater than for the second case, which shows that the manner of closing the window has an important influence on the leakage. It was also observed that improper application of paint at the cracks materially changed the leakage rate.

Tests on the felt weatherstripped window were made at three different humidities. For 100 per cent relative humidity, the air from the meter was assumed to be saturated. Practically dry air was obtained by the use of the

calcium chloride, while an intermediate humidity resulted from the use of calcium chloride not entirely dry. The humidities were measured by wet- and

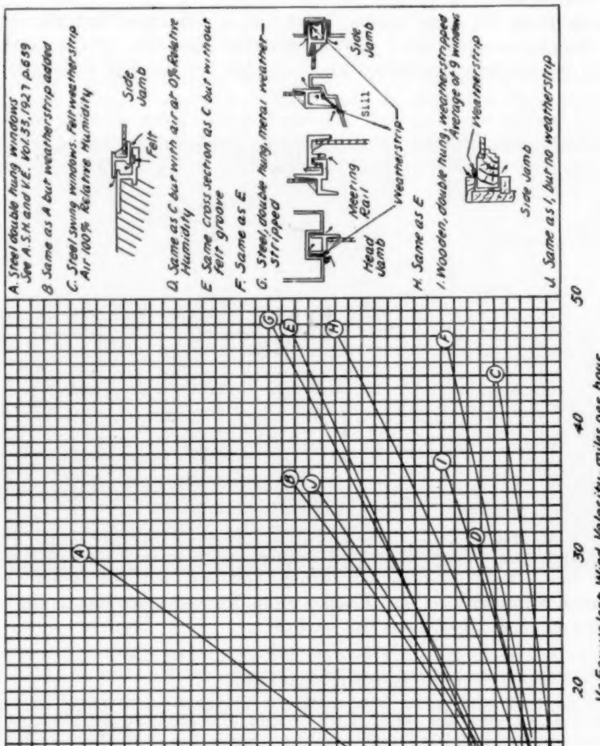


FIG. 3. INFILTRATION THROUGH VARIOUS KINDS OF WINDOWS

dry-bulb thermometers suspended in the hood. At each humidity, the flow of air for a particular pressure difference across the window was maintained until the infiltration rate became constant, which indicated that the moisture content

of the felt was approximately at a maximum (corresponding to the equilibrium moisture content).

The effect of humidity on leakage through felt weatherstripping, arranged as in Fig. 4, is shown in Fig. 6. Air leakage, in cubic feet per minute per linear foot of crack, is plotted as a function of equivalent wind velocity, in miles per hour. The air leakage at an equivalent wind velocity of 25 mph as a fraction of the leakage for saturated air is shown as a function of relative

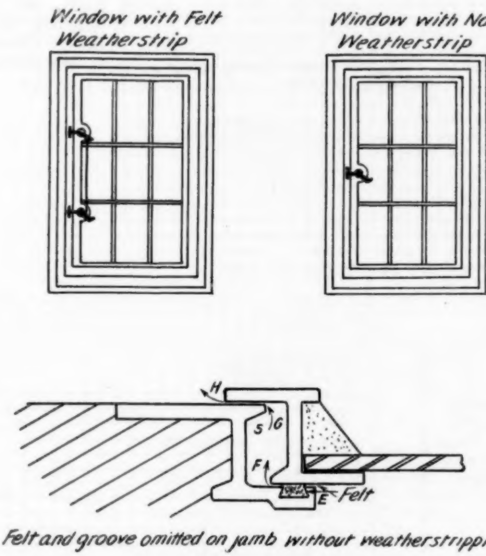


FIG. 4. METHOD OF MOUNTING WINDOWS IN HOODS AND CROSS SECTION OF JAMBS ON WEATHERSTRIPPED WINDOWS

humidity in Fig. 6a. It will be noticed that for this equivalent velocity, the leakage for dry air is 240 per cent of the leakage for saturated air.

If the air leakage per unit time as a function of pressure head across the window is plotted on logarithmic paper, the straight lines indicate a relation of the form:

$$Q = CAh^{a}$$

where

Q = quantity flowing in cubic feet per minute per foot of crack.
C = discharge coefficient.
A = effective area for flow through crack in square feet.
h = pressure head in inches of water.
a = exponent depending on the moisture content of the felt.

The exponent a was found to vary from 0.63 for dry air to 0.79 for saturated air. Since the leakage decreases with wet air, it is seen that the flow ap-

proaches that of the viscous type for the lower rates of flow, that is, higher humidities. In the viscous [5] region the leakage Q is proportional to the head

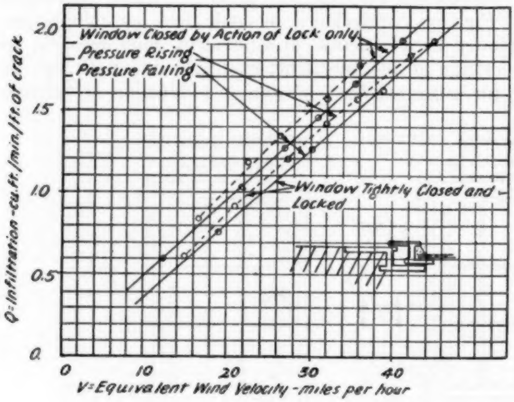


FIG. 5. EFFECT OF TIGHTNESS OF CLOSING OF WINDOW ON LEAKAGE OF AIR (WINDOW NOT WEATHERSTRIPPED)

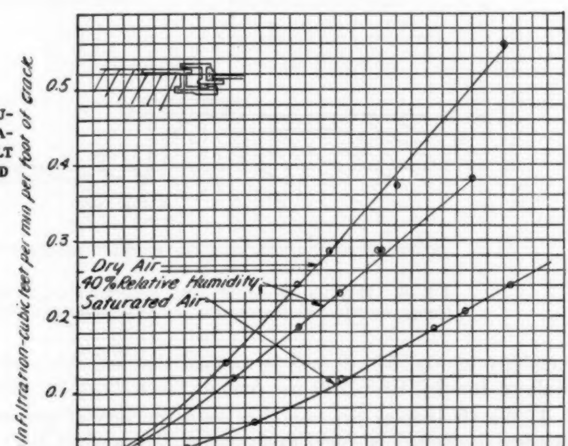


FIG. 6. EFFECT OF HUMIDITY ON INFILTRATION THROUGH FELT WEATHERSTRIPPED METAL WINDOWS

h while in the turbulent region Q is proportional to the square root of the head, $(h^{1/2})$.

Plotting $\frac{Q}{h^{n}}$ against the equilibrium moisture content of felt (obtained from Fig. 8) yields an approximately straight line as shown in Fig. 9. The moisture

[5] Gibson, Hydraulics and Its Applications, page 45.

content[6] of wool felt plotted in Fig. 8 was taken from the International Critical Tables, Vol. 2, pages 222-25.

The results plotted in Fig. 9 indicates that CA $(=\frac{Q}{h^a})$, the product of the discharge coefficient (C) and the cross sectional area (A) of the leakage path, and consequently the area (A), is influenced by the moisture content of the felt. This curve is plotted with but three points and therefore should only be interpreted as indicating a general trend.[7]

The leakage path is shown by the arrows in Fig. 4. The resistance to flow may be regarded as consisting of an entrance loss at E, friction loss through the path bounded by metal, friction loss through the felt, friction through the remainder of the first crack, expansion loss at F into space S, the friction loss

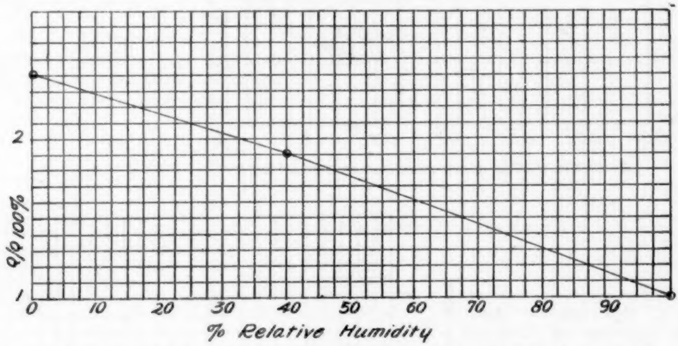


FIG. 6a. RATIO OF INFILTRATION OF PARTIALLY SATURATED AIR TO INFILTRATION OF SATURATED AIR THROUGH A FELT WEATHERSTRIPPED WINDOW. (EQUIVALENT WIND VELOCITY, 25 MPH)

due to flow through space S (which is small), entrance loss at G, friction loss through GH and exit loss at H. The change in the product of the coefficient of discharge (C) and the effective cross sectional area (A) of the leakage path for a given flow rate is partly due to the variable moisture content of the felt corresponding to various air humidities.

The actual variation in CA as the relative humidity is changed from 0 per cent to 100 per cent may be partially obscured by the decrease of frictional losses other than that through the felt. This would be due to the smaller infiltration rate and the consequent decrease in velocity through the leakage path. The loss in head due to sudden expansion[8] may be shown to vary with the square of the difference of the velocities before and after the expansion.

By means of the law of continuity it may be shown that the head loss is proportional to the square of the velocity through the smaller cross section. Similarly the head loss due to sudden contraction may be shown to be propor-

[6] See also Hougen and Watson, Industrial Chemical Calculations, page 168.

[7] Further tests are being conducted in which the diameter of a felt thread, the equilibrium moisture content, and the pressure drop through felt at various compression pressures are being determined for different humidities.

[8] Gibson, Hydraulics and Its Applications, pages 91 and 99.

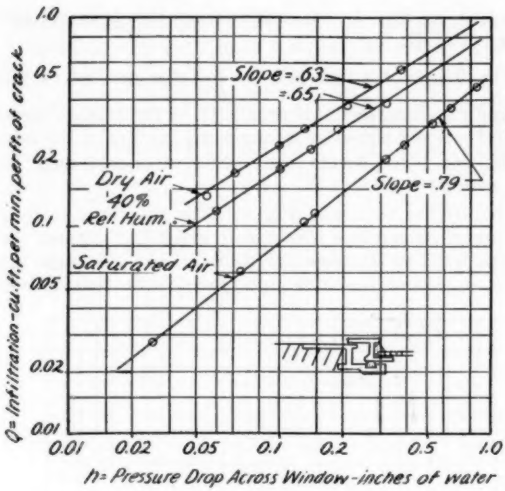


FIG. 7. EFFECTS OF HUMIDITY ON INFILTRATION THROUGH FELT WEATHER-STRIPPED METAL WINDOWS

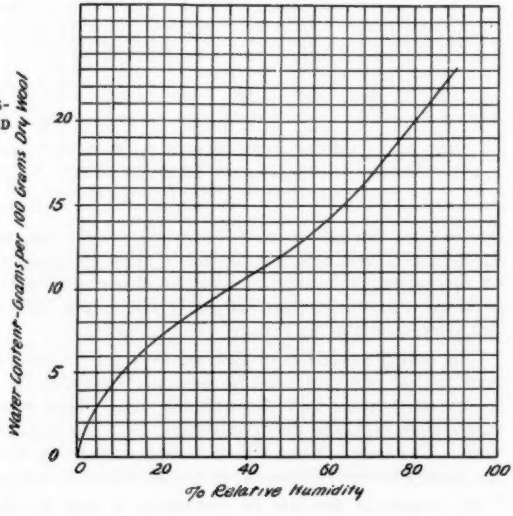


FIG. 8. EQUILIBRIUM MOISTURE CONTENT OF WORSTED WOOL AT 25 C *

tional to the square of the velocity in the smaller cross section. These two head losses (the latter being the smaller of the two) partly explain the deviation of the flow from the viscous flow law.

Replotting the equilibrium moisture content curve, Fig. 8, of wool felt in

* International Critical Tables, Vol. 2, pp. 222-225.

accordance with the method outlined by Bray and Draper,[9] assuming the relative humidity to be directly proportional to the partial pressure of the water vapor, yields Fig. 8a. This curve indicates that for relative humidities less than 50 per cent the phenomenon is one of adsorption, while above 50 per cent relative humidity, capillary condensation (absorption) occurs.

From Fig. 9 $\frac{Q}{h^a}$, that is the product CA, varies from 1.02 to 0.52. If C is assumed constant, then the variation of cross sectional area of flow, due to the change of moisture content of the felt, is 49 per cent. This would

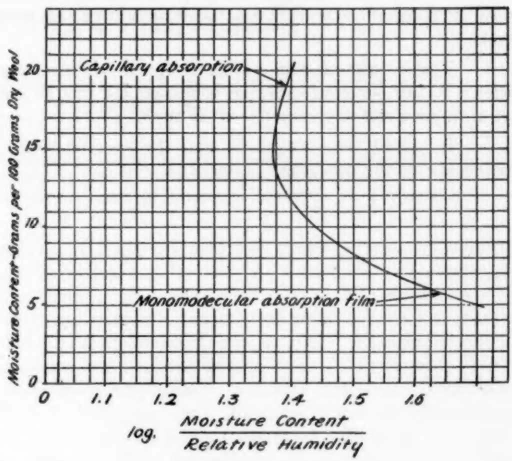


FIG. 8a. EQUILIBRIUM MOISTURE CONTENT OF WORSTED WOOL AT 25 C

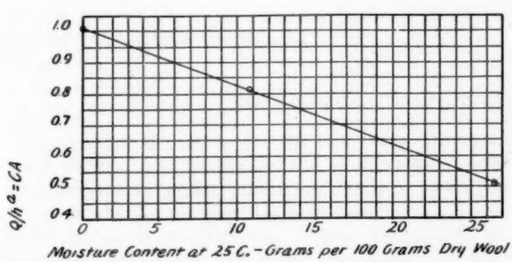


FIG. 9. EFFECT OF EQUILIBRIUM MOISTURE CONTENT ON FLOW OF AIR THROUGH WOOL FELT

represent the magnitude of the change of leakage rate, if the flow did not become more viscous as the area decreased. However, from Fig. 6, at an equivalent velocity of 25 mph, the air leakage rate falls from 0.48 to 0.20 cfm per foot of crack with the change from dry to saturated air. This decrease of 58 per cent is due both to the loss of area and change of the character of flow.

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[9] Capillary Condensation and Absorption, Bray and Draper, Proceedings, *National Academy of Sciences*, Vol. 12, No. 5, 1926.

CONCLUSIONS

1. A wide variation in the infiltration rate of well constructed windows may be expected. Fig. 3 shows variation of 1400 per cent at an equivalent velocity of 25 mph (compared to the smallest leakage shown).

2. Test results on windows differ widely depending upon the method of closing and latching. Variations of 15 per cent at 25 mph are indicated in Fig. 5.

3. The air leakage for steel swing windows is found to differ when determined with ascending and descending pressure differences because the window is closed more tightly after completion of the ascending pressure difference tests.

4. Test results on felt weatherstripped windows vary widely with air humidities. A 140 per cent increase is shown for dry as compared with saturated air at an equivalent air velocity of 25 mph.

5. It would seem desirable that the specifications for window tests include reference to the preceding Items 2, 3, and 4.

No. 953

FLOW OF CONDENSATE AND AIR IN STEAM-HEATING RETURNS

By F. C. Houghten *(*MEMBER*)* and Carl Gutberlet † (*NON-MEMBER*)
Pittsburgh, Pa.

This paper is the result of research conducted at the Research Laboratory of the American Society of Heating and Ventilating Engineers in cooperation with the Heating and Ventilating Department of Carnegie Institute of Technology

DATA covering capacities of 1-in. dry and wet return mains and return risers were recently published[1, 2] by the A. S. H. V. E. Research Laboratory. It was thought that the results of these studies would serve as a basis for revising the return capacities of pipe size tables in the A. S. H. V. E. Guide. However, in endeavoring to apply the data obtained to the development of capacities for returns larger than 1 in., it was found that there were still too many unknown factors, and as a result the Technical Advisory Committee on Pipe and Tubing Carrying Low Pressure Steam and Hot Water, of which S. R. Lewis is chairman, requested additional information on 1½-in. and 2-in. returns. The Laboratory was also requested to supply data on the rates of condensation and air return during the heating-up period from actual installations.

Test Equipment and Operation

In general, the set-up for the study and the methods of conducting the tests were similar to those used in the earlier studies on 1-in. pipe. Since an excessive amount of steam condensing surface would have been necessary for the tests on the larger pipe, it was decided to simulate air and water flowing from radiators through the return risers and mains by mixing measured quantities of air and hot water, and admitting this mixture to the top of the return riser. This obviated the necessity of supplying a great amount of

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* Director, Research Laboratory, American Society of Heating and Ventilating Engineers.
† Research Assistant, Research Laboratory, American Society of Heating and Ventilating Engineers.

[1] Capacity of Dry Return Mains for Steam and Vapor Heating Systems, by F. C. Houghten and Carl Gutberlet, A. S. H. V. E. Transactions, Vol. 36, 1930, p. 481.

[2] Capacity of Return Risers for Steam and Vapor Heating Systems, by F. C. Houghten and Carl Gutberlet, A. S. H. V. E. Transactions, Vol. 37, 1931, p. 189.

Presented at the 39th Annual Meeting of the American Society of Heating and Ventilating Engineers, Cincinnati, Ohio, January, 1933, by J. L. Blackshaw.

heating surface with an accompanying large steam cost, and worked well in operation, giving satisfactory control of all of the factors entering into the problem.

A diagrammatic sketch of the equipment used in the test set-up is shown in Fig. 1. Two different systems having the dimensions given in Table 1

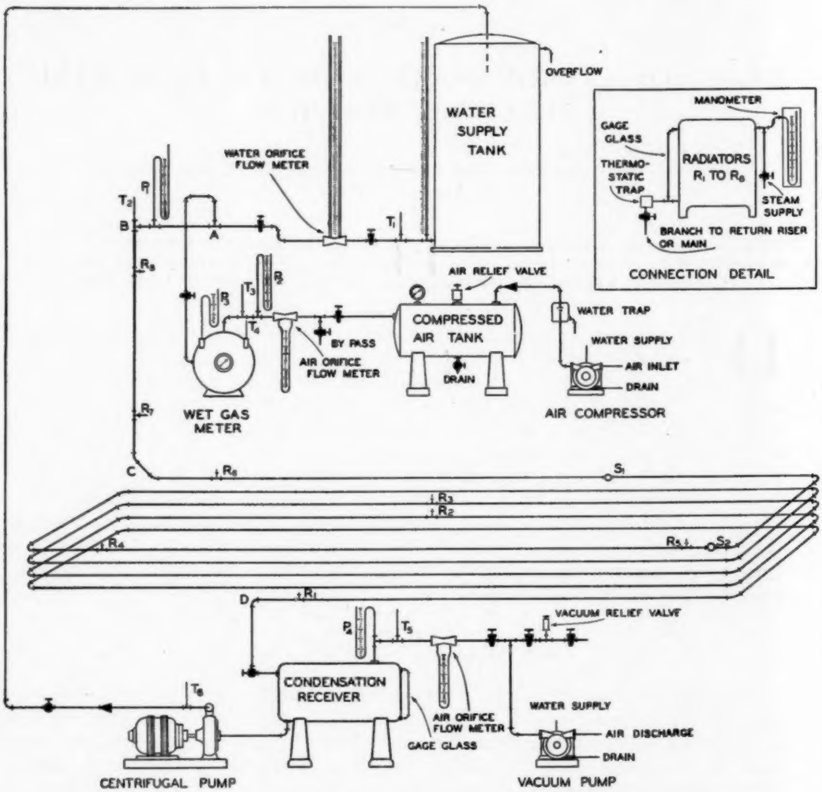

FIG. 1. LABORATORY SET-UP FOR STUDYING CAPACITIES OF RETURNS

were studied. Water at steam temperature was delivered at a constant rate from the 150-gal water supply tank through the water orifice flow meter and the horizontal pipe *AB* to the top of the return riser *BC*. At *A*, air was added to the horizontal pipe from the compressed air tank. The rate of supplying the air was controlled by the air orifice flow meter, but the volume added was actually measured at its existing temperature and pressure by the

wet gas meter. The 10-ft, 2-in. horizontal pipe *AB* was considered of sufficient length to allow the water and air to adjust themselves, so that by the time they reached *B* they would be in a condition as though they had come from heating units in a steam heating system. The air and water mixture, after passing down the return riser *BC* and through the coils of the return main *CD*, was delivered to the condensation receiver, from which, under gravity operation, the air was allowed to escape to the atmosphere, or from which it was removed by the vacuum pump. Water from the condensation receiver was

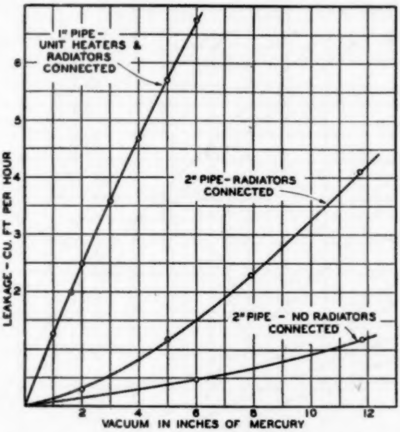

FIG. 2. LEAKAGE OF AIR INTO PIPE SYSTEMS

returned by the centrifugal pump to the water supply tank, where it was reheated by passing steam directly into it.

Two 45-deg elbows connected the return riser into the return main, which in each case consisted of about 319 linear feet of the pipe tested. The return main was fitted into four complete loops when the 1½-in. pipe was used, and five complete loops when the 2-in. pipe was used. It was accurately laid out with a pitch of 1 in. in 16 ft. The ends of the loops were 18 in. long.

Along the return riser and main, eight radiators, each of 12 sq ft (equivalent) capacity, were connected at points R_1 to R_8 as shown in the sketch. These

TABLE 1. DIMENSIONS OF THE PIPE SYSTEMS STUDIED

	Size of Pipe		Distance from Lower End of Main in Feet									
	Riser	Main	R_1	R_2	R_3	R_4	R_5	R_6	R_7	R_8	Bottom of Riser	Top of Riser
System 1	1½	1½	1.75	96.30	178.75	242.90	279.20	316.20	321.00	329.20	319.18	330.78
System 2	1½	2	2.54	105.35	173.41	260.98	289.98	315.47	320.03	328.38	318.45	329.96

radiators served to indicate when the return at any particular point was overloaded. With the steam supply valve to the radiator closed, and the gate valve between the thermostatic trap and the return open, the manometer on the radiator indicated the pressure in the return.

In using a radiator to determine when the return was overloaded, the valve

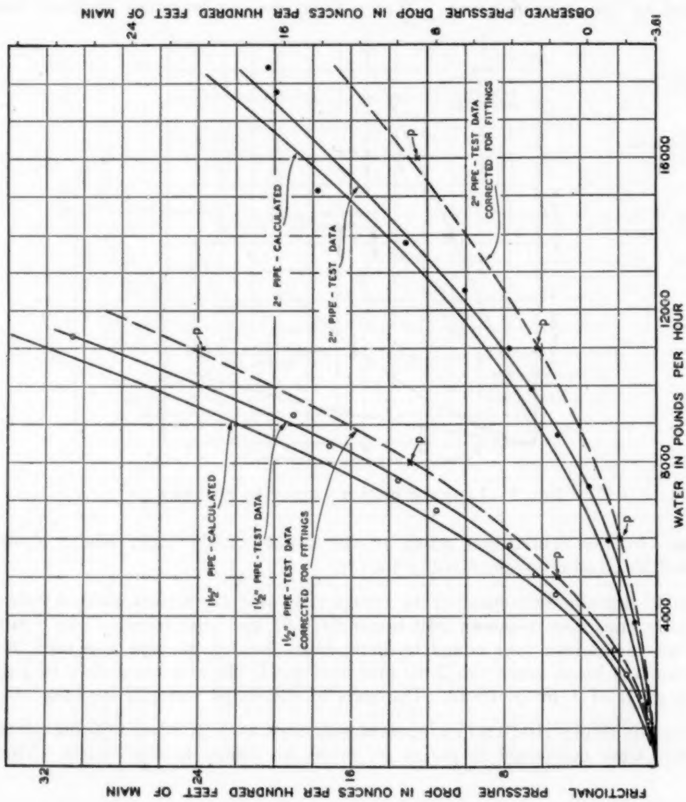

FIG. 3. FLOW OF WATER THROUGH WET RETURN MAINS

between the radiator and the return was left open, while the steam supply valve was opened and adjusted to maintain a pressure in the return ¾ oz greater than that observed before the steam was admitted. If any radiator required more than six minutes to heat up, the return at this point was considered to be overloaded, basing this time on the assumption that a fairly large radiator, of about 60 sq ft (equivalent) capacity, should heat up in thirty minutes for satisfactory operation. Actually, it was found that if the pressure

in the return was 1 oz or less before admitting steam, the radiator heated up satisfactorily within the accepted time, while if the pressure in the return was greater, the radiator seldom heated up in this time.

To observe the effect of fittings on the pressure drop, the radiators were connected to the return mains so that pressure drops could be obtained through the entire length of the return, through one straight length of pipe, or through one complete loop. It was found, however, that pressure drops could not be determined accurately enough through such short sections to furnish satisfactory data on this point. In the system, other pressures were observed by manometers at points P_1 to P_4. Temperatures were observed by thermometers at points T_1 to T_5. Through glass observation sights located in

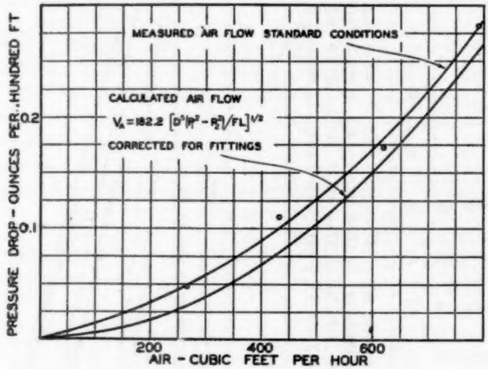

Fig. 4. Flow of Air Through the 2-In. Main

tees at S_1 and S_2, the degree of turbulence of flowing water could be observed in either a straight run of pipe or immediately after the water had flowed through an elbow.

Test Results

With the set-up as described, great freedom was possible in obtaining data. Any reasonable amount of air and water in any ratio could be supplied at the top of the system, while the bottom could be operated under atmospheric conditions or a wide range of vacuum. By simple mechanical changes of the equipment, it was possible to measure the air leakage into the system under varying degrees of vacuum, to measure the characteristics of hydraulic flow when the main was operating full of water as in a wet return, and to study the flow characteristics of air.

Air Leakage Into the System

Before any runs were made on the system, it was tested for tightness by maintaining definite and constant vacuums and measuring the volume of air

TABLE 2. TYPICAL TEST DATA

Test No.	Pipe Size, In.	Barometric Pressure	Size H_2O Orifice, In.	Size Air Orifice, In.	Diff. Press. H_2O Orifice In. H_2O	Rate H_2O Lb per Hour	Diff. Press. Air Orifice In. H_2O	Air Rate Cu Ft Air per Hour	Temperature—Deg Fahr					Gage Pressures—In. Hg										Press. Drop Oz per 100 Ft
									Supply H_2O	Top of Riser	Return H_2O	Air Orifice	Air Meter	Air Orifice	Air Meter	Horizontal Main						Riser		
									T_1	T_2	T_6	T_3	T_4	P_2	P_8	R_1	R_2	R_3	R_4	R_5	R_6	R_7	R_8	
31	1½	28.97	0.50	0.25	11.7	1570	3.2	100.98	207.5	198	174	95.00	95.50	+.30	+.27	−1.55	− .90	− .62	−.22	0.0	+.35	+.28	+.23	4.24
52	1½	29.21	0.50	0.25	11.7	1570	28.0	297.00	207.0	194	169	86.00	86.00	+.38	+.35	−3.00	−2.20	−1.30	−.40	−.30	+.15	+.15	+.20	8.16
70	2	28.99	0.25	0.25	18.00	800	30.0	303.6	207.0	178	146	89.25	85.50	+.20	+.16	− .02	+ .02	+ .03	+.06	+.06	+.07	+.09	+.10	0.21
101	2	29.20	0.75	0.25	17.75	4000	7.0	143.6	208.0	203	189	88.25	88.25	+.90	+.85	−4.00	−2.80	−1.70	−.50	−.15	+.25	+.40	0.0	10.83

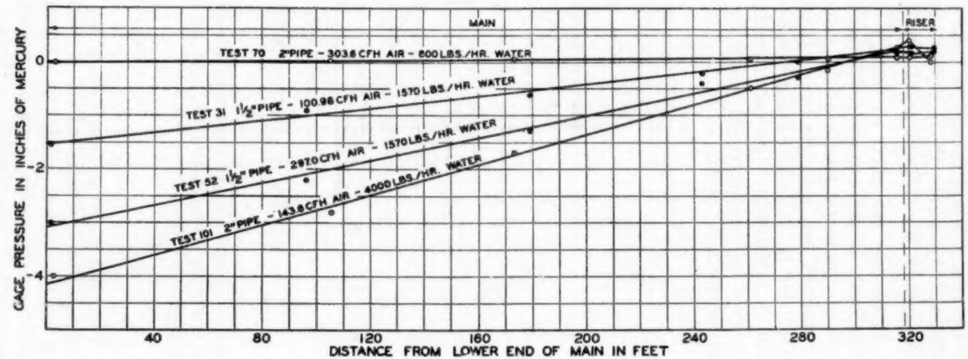

FIG. 5. PRESSURE DROP FOR FLOW OF WATER AND AIR THROUGH PIPE

exhausted in holding these vacuums. These air volumes were determined with the pipe capped at the top of the riser, first with the eight test radiators connected, and second with them disconnected and their return openings plugged. The leakage of air into the system for the 2-in. main, both with and without radiators and fittings, is given in Fig. 2, where the curves show leakage against vacuum maintained for the two conditions, and also the leakage found in the earlier study [1] on a similar set-up with 1-in. pipe. Contrary to expectations, there is considerably more leakage in the case of the 1-in. pipe than with the 2-in., but this may be explained by the greater number of fittings used in the 1-in. set-up, where all water carried was actual condensation from unit heaters. It should be emphasized that the air leakage tests were made under conditions which would give a larger value than would result in actual systems operating under natural conditions. The pipe was cold and the vacuum indicated was applied to the entire length of the system instead of at the pump with a gradually diminishing vacuum from this point to the most remote radiator. Under a vacuum of 12-in. of mercury, the 2-in. pipe system leaked 4.2 cu ft per hour when the radiators were connected, and 1.2 cu ft per hour when they were not connected. It was found in the tests on the dry return main that the 2-in. pipe operating under a pressure drop of 1 oz per 100-ft length of run and carrying 0.1 cu ft of air per pound of water, or a rate of air delivery equivalent to an average heating-up period, would carry 189 cu ft of air per hour. Hence, the significance of the quantity of air found leaking into the system on its capacity is small. The leakage definitely shows, however, that workmanship and the number of fittings are exceedingly important in the erection of piping where high vacuums are to be maintained.

Wet Return Mains

By excluding air, filling the entire pipe system with water, and controlling the flow by the valve between the main and the condensation receiver, the flow pressure drop relation for the main acting as a wet return was determined, see results for the 1½-in. and 2-in. mains, Fig. 3. Pressure drop curves are shown for the test data using the actual length of pipe including fittings, for the values from these curves corrected for fittings, and for the relations as calculated by the following generally accepted hydraulic formula (including a correction for fittings):

$$W = 3{,}600\ DAC\sqrt{RS} \qquad (1)$$

where

W = weight of water in pounds per hour
D = density of water in pounds per cubic foot
C = a constant, depending upon the velocity of flow and smoothness of pipe
R = mean hydraulic radius
S = pressure drop in feet of water column per linear foot of pipe

In using Formula 1, the choice of values for C is important. The calculated curve was drawn by using C equal to 79.8, 81.2, and 82.2 for velocities corresponding to 4,000, 8,000 and 11,000 lb of water per hour for the 1½-in. pipe, and C equal to 82.5, 86.0 and 88.8 for velocities corresponding to 6,000, 12,000 and 18,000 lb of water per hour for the 2-in. pipe, as given by Marks Engineers

TABLE 3. RELATION OF PRESSURE DROP AND AIR-WATER RATIO TO CONDENSATE LOAD FOR DRY RETURN MAINS

Nominal Size of Pipe—Inches		¾	1	1¼	1½	2	2½	3	3½	4
Pressure Drop Oz per 100 Ft *P*	Cu Ft Air per Pound Water *R*	Condensate Load Pounds per Hour								
0.5	.01	183	425	886	1282	2247	3296	5201	7025	9106
1		280	557	1084	1537	2639	3837	6014	8099	10477
2		402	722	1333	1856	3131	4517	7036	9447	12198
3		488	839	1507	2080	3477	4995	7753	10394	13406
4		557	931	1646	2259	3752	5375	8325	11148	14368
5		615	1010	1764	2410	3985	5697	8809	11787	15184
10		825	1296	2192	2961	4835	6871	10572	14114	18154
0.5	.02	139	366	798	1169	2072	3054	4838	6546	8494
1		224	481	970	1390	2413	3525	5546	7480	9686
2		330	624	1185	1667	2840	4114	6431	8648	11577
3		404	725	1336	1860	3138	4526	7050	9466	12221
4		463	805	1456	2014	3376	4855	7543	10116	13052
5		513	872	1557	2145	3577	5132	7960	10667	13754
10		695	1118	1926	2619	4307	6142	9477	12668	16309
0.5	.03	114	331	746	1102	1970	2912	4625	6265	8135
1		191	436	903	1304	2281	3342	5271	7118	9224
2		287	567	1099	1556	2669	3878	6077	8180	10580
3		355	658	1236	1732	2940	4253	6639	8923	11528
4		408	730	1345	1871	3155	4550	7086	9513	12282
5		454	792	1437	1990	3337	4802	7464	10012	12918
10		618	1014	1770	2418	3999	5715	8836	11823	15230
0.5	.04	96	307	710	1055	1897	2812	4474	6066	7881
1		168	405	856	1243	2187	3212	5077	6860	8896
2		257	526	1038	1477	2548	3711	5825	7849	10157
3		320	611	1165	1640	2799	4059	6347	8538	11037
4		369	678	1266	1770	2999	4334	6762	9085	11735
5		411	735	1350	1879	3168	4567	7112	9547	12325
10		563	941	1660	2276	3780	5413	8382	11223	14464
0.5	.05	82	288	681	1018	1840	2734	4357	5911	7684
1		150	380	819	1196	2114	3112	4925	6661	8641
2		234	494	991	1416	2454	3581	5630	7591	9829
3		293	574	1110	1570	2790	3908	6121	8239	10655
4		339	637	1205	1691	2878	4167	6510	8753	11311
5		378	690	1285	1794	3036	4386	6839	9186	11865
10		521	883	1574	2166	3710	5178	8029	10758	13870
0.5	.06	70	272	658	988	1794	2670	4262	5785	7523
1		135	360	789	1157	2055	3030	4802	6498	8433
2		215	469	952	1366	2377	3475	5471	7381	9560
3		271	544	1065	1512	2601	3785	5936	7995	10344
4		315	604	1155	1627	2779	4030	6305	8481	10965
5		352	654	1230	1724	2929	4237	6615	8892	11489
10		487	837	1504	2076	3471	4986	7741	10377	13385

TABLE 3. RELATION OF PRESSURE DROP AND AIR-WATER RATIO TO CONDENSATE LOAD FOR DRY RETURN MAINS—*Continued*

Nominal Size of Pipe—Inches		¾	1	1¼	1½	2	2½	3	3½	4
Pressure Drop Oz per 100 Ft *P*	Cu Ft Air per Pound Water *R*	Condensate Load Pounds per Hour								
0.5	.07	60	259	638	963	1755	2611	4180	5678	7386
1		122	343	764	1125	2005	2960	4698	6361	8257
2		199	447	919	1324	2312	3385	5336	7203	9333
3		252	519	1027	1463	2526	3681	5780	7789	10080
4		294	576	1112	1573	2695	3914	6131	8252	10672
5		329	624	1184	1665	2838	4111	6427	8643	11171
10		458	797	1445	2000	3354	4824	7340	10056	12975
0.5	.08	52	248	621	941	1722	2570	4111	5586	7269
1		112	328	741	1097	1961	2900	4607	6241	8105
2		185	428	891	1288	2256	3308	5220	7050	9137
3		236	497	994	1421	2461	3590	5644	7610	9852
4		276	551	1076	1526	2622	3814	5980	8053	10418
5		310	597	1145	1614	2759	4003	6263	8427	10895
10		432	763	1393	1934	3252	4684	7287	9778	12619
0.5	.10	38	229	593	905	1665	2492	3994	5431	7071
1		93	304	705	1049	1888	2800	4456	6042	7851
2		161	396	843	1227	2185	3178	5025	6792	8808
3		209	460	939	1350	2252	3440	5418	7311	9471
4		246	510	1015	1447	2501	3647	5729	7721	9994
5		277	553	1078	1529	2627	3821	5990	8067	10435
10		390	706	1308	1824	3082	4449	6934	9312	12026
0.5	.15	12	194	541	838	1563	2350	3781	5151	6713
1		61	259	638	964	1756	2617	4182	5680	7388
2		119	339	757	1116	1991	2942	4671	6325	8212
3		159	393	839	1221	2153	3166	5007	6768	8778
4		191	436	903	1304	2281	3342	5271	7118	9295
5		218	472	958	1374	2388	3490	5494	7412	9599
10		313	602	1152	1624	2773	4023	6294	8467	10946
0.5	.20		170	504	791	1490	2250	3630	4952	6459
1		37	228	591	903	1662	2487	3987	5423	7060
2		89	298	696	1038	1870	2775	4419	5993	7788
3		124	346	768	1130	2013	2972	4715	6383	8286
4		152	384	825	1203	2125	3126	4947	6690	8678
5		175	415	872	1264	2218	3256	5142	6947	9006
10		259	528	1041	1482	2555	3720	5839	7867	10181
0.5	.25		151	476	755	1434	2172	3513	4797	6262
1		19	203	554	856	1589	2387	3836	5223	6806
2		66	266	649	977	1777	2645	4224	5736	7460
3		97	309	713	1059	1904	2821	4489	6085	7905
4		122	343	763	1124	2003	2959	4695	6358	8254
5		143	371	806	1178	2087	3074	4869	6586	8546
10		217	471	956	1371	2385	3485	5487	7402	9587

TABLE 3. RELATION OF PRESSURE DROP AND AIR-WATER RATIO TO CONDENSATE LOAD FOR DRY RETURN MAINS—*Continued*

Nominal Size of Pipe—Inches		¾	1	1¼	1½	2	2½	3	3½	4
Pressure Drop Oz per 100 Ft P	Cu Ft Air per Pound Water R	Condensate Load Pounds per Hour								
0.5	.30		135	453	725	1388	2108	3418	4671	6101
1			183	524	817	1530	2305	3713	5060	6598
2		47	240	610	927	1700	2539	4065	5525	7191
3		75	279	668	1002	1815	2698	4304	5841	7594
4		97	309	713	1060	1904	2822	4480	6086	7907
5		116	335	751	1108	1979	2926	4646	6292	8170
10		183	424	886	1281	2246	3294	5199	7022	9101

Handbook.[3] It will be observed that the calculated curve gives considerably lower rates of water flow for the same pressure drop than found by test when both are based upon corrected lengths of pipe. This discrepancy may result from error in the accepted correction for length of pipe for fittings, or it may more likely result from incorrect values of C. The values of C used in the formula were determined many years ago, and may apply for pipe with a considerably rougher internal surface than that found on the market at the present time. It is well known that improvements in the manufacture of pipe have resulted in smoother surfaces. The pipe used in the set-up was the average run of steel pipe found on the market.

If the discrepancies between the calculated and actual flow curves, both corrected for fittings, result from erroneous values of C, then it is of interest to see how much the constant C must be changed in order to make the formula approximate more closely the actual flow in modern pipe. The points P for both sizes of pipe were calculated from the formula but with the accepted values of C from Marks changed by the factor 1.245. It will be observed that these points fit the test data corrected for fittings very well for rates of flow up to about 10,000 lb of water per hour for the 1½-in. pipe and up to 14,000 lb of water per hour for the 2-in. pipe. Above these rates of flow, the formula with the corrected C appears to give rates of flow slightly greater, which may result from errors in experimental data above these rates of flow where few data were taken.

The pressure drops due to the frictional resistance to flow are given by the scale on the left hand side of the chart. Actual pressures, measured at points R_1 to R_8 by manometers with one leg open to the atmosphere, were used in calculating the apparent pressure drops which are shown by the scale at the right of the chart. This scale applied only to the curves from test data. For the condition of no flow the observed pressure drop is the hydrostatic head, which based on a 100-ft length of the main, pitched 1 in. in 16 ft, is 3.61 oz.

[3] *Marks Handbook,* Third Edition, p. 275.

For conditions of flow the pressure drop due to the frictional resistance in the pipe is the observed pressure drop plus the hydrostatic head of 3.61 oz per

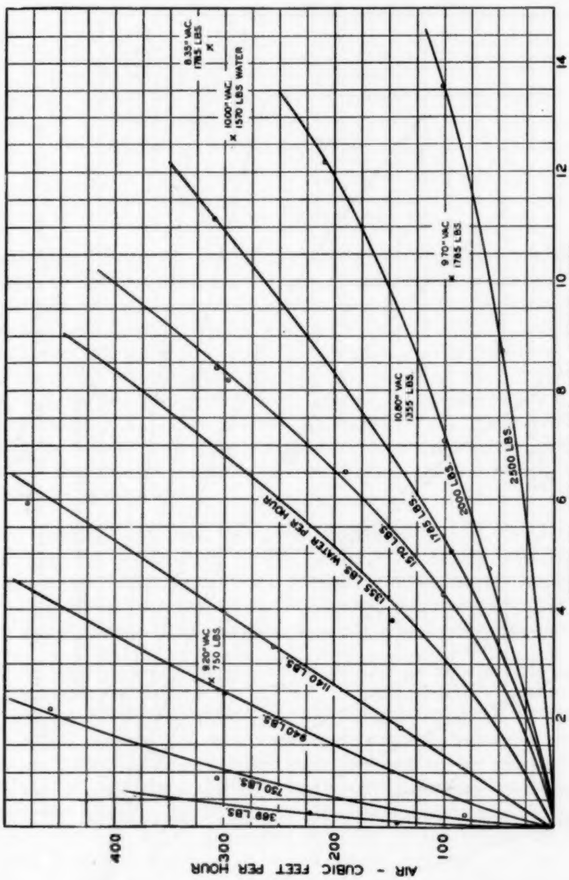

Fig. 6. Relation of Pressure Drop to Flow of Air and Water Through 1½-In. Pipe

100 ft. It will be noted that with no observed pressure drop as indicated by the scale on the right there was considerable flow through both sizes of pipe. This represents a condition where the frictional resistance to flow just equals the hydrostatic head resulting from the pitch of the pipe.

Flow of Air in Return Mains

A few tests were made with only air flowing through the returns. The

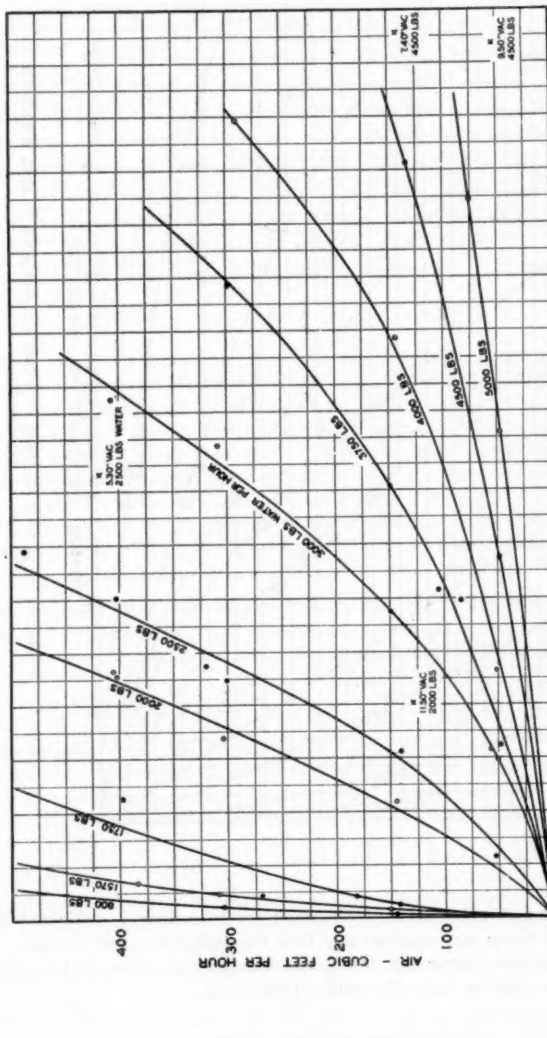

FIG. 7. RELATION OF PRESSURE DROP TO FLOW OF AIR AND WATER THROUGH 2-IN. PIPE

results are plotted in Fig. 4. A curve is also plotted based upon the following formula and constants given by Marks[4]:

$$V_a = 182.2\ [d^5(p_1^2 - p_2^2)\ /\ fL]^{1/2} \qquad (2)$$

where

V_a = cubic feet of air per hour at 60 F and 29.92 in. of mercury
d = internal diameter of pipe in inches
p_1 and p_2 = initial and final pressures in pounds per square inch
L = length of pipe in feet
f = coefficient of friction $= 0.0028\left(1 + \frac{3.6}{d}\right)$

DRY RETURN MAINS

Tests were run on the system under a wide range of practical conditions of operation to enable the characteristics of dry returns to be studied. The rates at which air and water were allowed to flow through the system were predetermined and held constant by the valves between *A*, Fig. 1, and the compressed air tank and the water flow orifice, respectively. For each test using vacuum, the pressure in the condensation receiver was adjusted for the predetermined water load, so that the pressure drop through the system would be just sufficient to allow radiator R_6, farthest from the vacuum pump, to be heated up in about six minutes. Tests with gravity operation were made under similar conditions, excepting that the air entering the receiver from the return was allowed to escape to the atmosphere.

After conditions of test had been established and all factors maintained in equilibrium for a sufficient length of time to obtain uniform performance throughout the entire system, all pressures, temperatures, and gage readings were observed and the rate of air supply was determined by the wet gas meter over a five-minute period. Typical data obtained by test are shown in Table 2.

All air volumes were corrected to saturated air at a pressure of 29.92 in. of mercury and a temperature of 85 F. This temperature was used rather than 60 F because it was found to be approximately a mean of the temperatures under which the wet gas meter operated in the entire series of tests. Unless otherwise stated, all air volumes are assumed to be measured under these conditions, though they are not the conditions under which the air flowed through the pipe, in which it was necessarily saturated at the water temperature. In Fig. 5, the pressures observed in the mains for the tests listed in Table 2 are plotted against actual length of pipe without correction for fittings. It was found that these curves were best drawn as straight lines from which the pressure drop for any test was determined in ounces per 100-ft length of pipe.

By plotting the volume of air handled in tests run with varying air rates at certain definite condensation loads against pressure drops through the main, the curves shown in Figs. 6 and 7 were drawn for the 1½-in. and the 2-in. dry return mains, respectively. The actual points on which the curves were based

[4] *Marks Handbook,* Third Edition, p. 376.

are shown. These curves are similar to those reported in the earlier paper on 1-in. pipe and show a tendency to reverse their direction for loads between

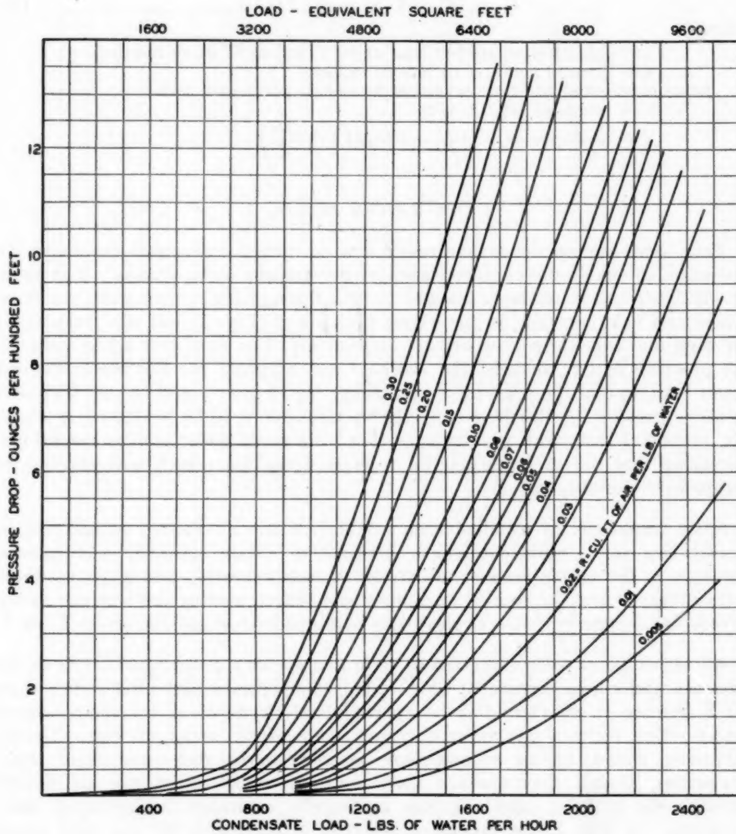

FIG. 8. RELATION BETWEEN PRESSURE DROP AND CONDENSATE LOAD FOR VARIOUS AIR-WATER RATIOS—1½-IN. PIPE

1,140 and 1,355 lb of condensate per hour for the 1½-in. pipe, and for loads between 1,750 and 2,000 lb for the 2-in. pipe. In order to study this change in characteristic of flow, the glass observation sights S_1 and S_2, Fig. 1, were installed, and it was found that for approximately the same load limitations the movement of water changed from comparatively quiet flow, with small waves in the lower part of the pipe, to turbulent flow, with water waves reaching the top of the pipe and with slugs of water separated by air pockets

pulsing through the pipe. This change in condition of flow no doubt accounts for the change in curvature.

In order to put the data from Figs. 6 and 7 in adequate form for building up

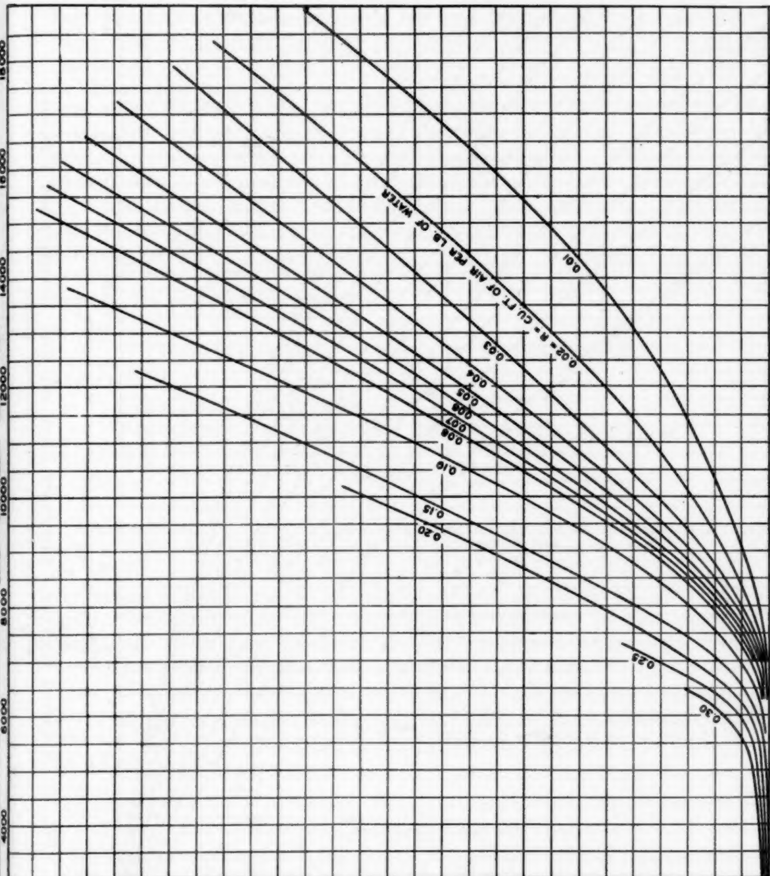

FIG. 9. RELATION BETWEEN PRESSURE DROP AND CONDENSATE LOAD FOR VARIOUS AIR-WATER RATIOS—2-IN. PIPE

tables of pipe sizes for use in design under any accepted condition of operation, the ratio of air in cubic feet per hour to water handled in pounds per hour was plotted against the water load and its accompanying pressure drop in Figs. 8 and 9, which give a series of curves for ratios ranging from 0.005 to 0.30 cu ft of air per hour per pound of water per hour.

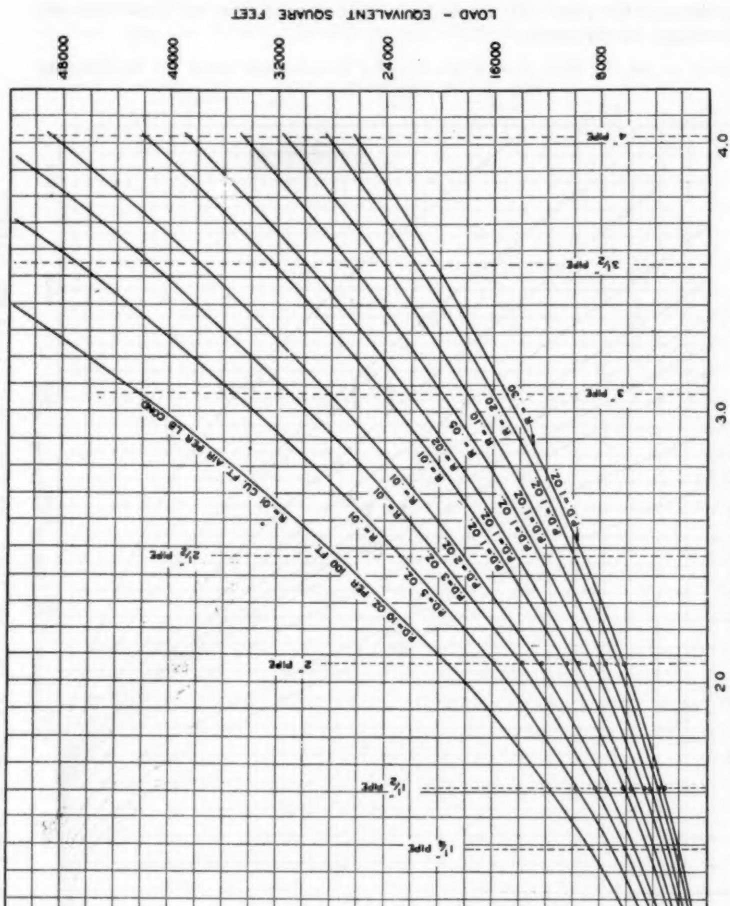

FIG. 10. RELATION BETWEEN CONDENSATE LOAD AND PIPE DIAMETER FOR VARIOUS AIR-WATER RATIOS AND PRESSURE DROPS

The capacity of return piping must be based upon the quantity of air and water to be handled when the two combine in some ratio to offer the greatest load on the system. This will obviously come at some time during the heating-up period, and will of necessity bear some relation to it. Hence, to accurately determine the proper size of a dry return main, it is necessary to know not only the heating-up period and load, but also either the maximum rate of air flow or the ratio of air to condensate. Therefore, Fig. 9 in the earlier report and Figs. 8 and 9 of this report may be used directly for design data, or design data may be prepared from them on any assumed ratio of air to condensate handled during the heating-up period. In order to be able to extrapolate the results of the study on these three sizes of pipe to larger sizes, curves are given in Fig. 10 for a 1-oz pressure drop with air-water ratios of 0.01 to 0.3, and also for an air-water ratio of 0.01 with pressure drops of 3, 5 and 10-oz. Similar series of curves were drawn for pressure drops varying from ½ to 10 oz, and equations applying to each band of such curves for each particular pressure drop were analyzed by plotting their component parts on semi-logarithmic and logarithmic paper, with the resultant development of the empirical formula,

$$L = (2.18\, P^{\,0.04820} - P^{\,0.36808} \log R) \times (157.08\, d^2 + 80) - 500 \qquad (3)$$

where

L = condensation load in pounds per hour
P = pressure drop in ounces per 100 feet
R = ratio—cubic feet of air per pound of condensate
d = actual internal diameter of pipe in inches

Values given in Table 3 are based upon Formula 3, which fits the actual test data to within ± 7 per cent, and are selected for a range of conditions on which any logical set of pipe size tables might be based. Pipe size tables for design purposes may be based upon curves similar to Fig. 10, Table 3, or upon Formula 3, after such operating characteristics as the length of the heating-up period, the maximum air and condensate return load during this period, the allowable pressure drop, and a possible factor of safety are agreed upon. To help establish these operating characteristics, the A. S. H. V. E. Research Laboratory has made a study of the rates of returning condensate and air in a few actual systems, the results of which will be made the basis of a later report.

Gravity Operation

Figs. 6 and 7 include data for both gravity and vacuum pump operation. Where gravity operation was possible and as long as the vacuum applied by the pump was small or just sufficient to overcome friction through the main, the pressure drop relation was the same for both types of operation. Gravity operation was possible only for comparatively small air and condensate loads, the limit in such loads being given in Fig. 11 for the 1½-in. and 2-in. pipe tested in this study and for the 1-in. pipe studied earlier. The limitation curve for a 1¼-in. pipe is drawn by interpolating between the curves for the pipes

tested. Air-water ratios as used in analyzing the vacuum pump data are also plotted, and in order to show the relation of air-water ratio to assumed

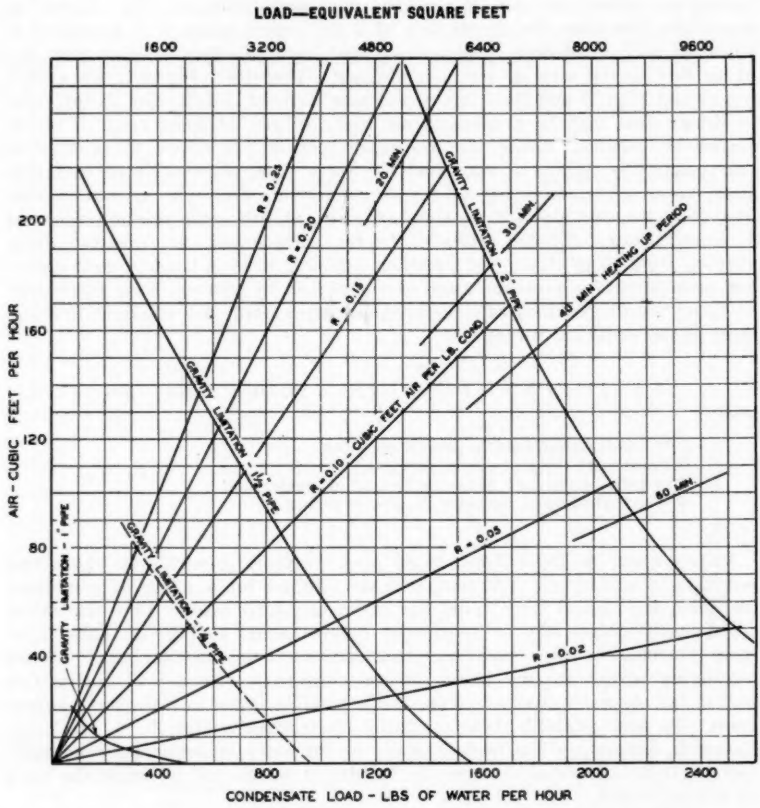

FIG. 11. LIMITING CAPACITY OF DRY RETURN MAINS FOR GRAVITY OPERATION

heating-up period as used in the 1-in. pipe study, the location of these heating-up period curves in relation to the ratio curves is indicated.

RETURN RISER

The return riser was approximately 11.5 ft in length with slightly more than 8 ft between the two radiators R_7 and R_8. This distance was too short to give a measurable pressure drop for accurate analysis, especially when the small pressure difference was masked by the large hydrostatic head of the water

in the riser. Hence, it is not possible as a result of the study to give accurate pressure drop flow data for the riser. Since the capacity of the 1½-in. riser was much larger than that of the 2-in. main, it was left in place while the 2-in. main was being studied. As nearly as could be observed, the capacity of the 1½-in. riser was equal to that of the 2-in. main, both for gravity and vacuum pump operation. For gravity operation, however, more erratic results were found for the riser than for the main. The riser occasionally operated satisfactorily at much greater loads than shown as the limiting capacity for the 2-in. main in Fig. 11. However, these greater capacities could not be depended upon, and as a result, it was concluded that the limiting capacity of the 1½-in. riser for gravity operation is the same as that for the 2-in. main.

HIGH VACUUM

The data presented in the curves in Figs. 6 and 7 and subsequently analyzed cover conditions of operation where the vacuum applied by the pump at the lower end of the main was just sufficient to overcome the frictional resistance to flow throughout the entire length of pipe. A few tests were made with a much higher vacuum which brought the entire return below atmospheric pressure. The results of these tests are shown as points indicated by X in Figs. 6 and 7, where the vacuum applied at the condensation receiver, in inches of mercury, and the rate of water carried, in pounds per hour, are listed beside such points. It will be seen that the pressure drop for the same water and air load is greatly increased. This is because of the increase in volume of air flowing under the lowered pressure, and to a lesser extent by the air leaking into the system, both of which took place after the volume of air as plotted was measured by the wet gas meter at atmospheric pressure.

SUMMARY

1. Leakage of air into piping systems under vacuum varies widely with the different set-ups studied, becoming greater as the number of fittings and valves around radiator connections increases. The total leakage, however, even with vacuums applied uniformly throughout the system larger than customarily used in practice, corresponds to comparatively small rates of air flow during normal operation of the system.

2. The pressure drop relation for a return main carrying only water, thus simulating a wet return main, is given for the system studied. Greater rates of water delivery were found for a given pressure drop than would be given by the usual hydraulic formula and accepted constants. Corrected constants giving flows equal to that observed are presented.

3. A much-used formula for flow of air in pipe is shown to give flows which agree closely with those experimentally determined.

4. The pressure-drop-flow relation is given for different ratios of air and water flowing in 1-in., 1½-in., and 2-in. pipe, when operating as a dry return main. The capacities of these three sizes of pipe when plotted against pipe

diameter give consistent curves, which are extrapolated with fair accuracy to larger pipe sizes.

5. A formula is developed giving the capacity of pipe for carrying condensate and air based upon the diameter of the pipe, the pressure drop, and the ratio of air to condensate carried. This formula, with curves from which it was developed and related tables, can be used for developing pipe size tables, when the flow characteristics in a heating system under practical conditions are agreed upon.

No. 954

CONDENSATE AND AIR RETURN IN STEAM HEATING SYSTEMS

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This paper is the result of research conducted at the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the Heating and Ventilating Department of Carnegie Institute of Technology

DATA giving the relation between pressure drop and the rate of flow of water and air in pipe, covering a wide range of percentages of air to water for different sizes of pipe, were published [1] recently by the A. S. H. V. E. Research Laboratory. These data may serve as the basis of the capacities of return piping in steam heating systems, but before developing practical pipe size tables, it was considered desirable by the Technical Advisory Committee on Pipe and Tubing Carrying Low-Pressure Steam and Hot Water, of which S. R. Lewis is Chairman, that the characteristics of operation of actual heating systems be studied under practical operating conditions.

The results of the former paper [1] indicate that at any given pressure drop the capacity of a pipe for carrying either air or water is much greater than its capacity for carrying a mixture of both. Hence, the greatest demand on the capacity of the returns of a steam heating system will come sometime during the heating-up period when they must carry more than the normal amount of air eliminated from the heating units and pipe and when the condensation rate is greater than during continued operation.

This paper reports the results of a study of the returning air and condensate from two actual heating systems during and after heating up, with both gravity and vacuum pump operation. One study was made on a two-pipe gravity vapor system in a 12-room residence containing 640 sq ft (equivalent) of direct radiation; the other, on that part of the vacuum pump differential system in the Grant Building, Pittsburgh, on the 34th to the 39th

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[1] Flow of Condensate and Air in Steam-Heating Returns, by F. C. Houghten and Carl Gutberlet, A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933; p. 179.

Presented at the 39th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Cincinnati, Ohio, January, 1933, by J. L. Blackshaw.

floors, which contained 2,510 sq ft (equivalent) of concealed, extended surface copper radiation.

Test Set-up and Operation

The pipe system and the arrangement of the test equipment for gravity operation in the residence are shown in Fig. 1, in which the size of the pipe

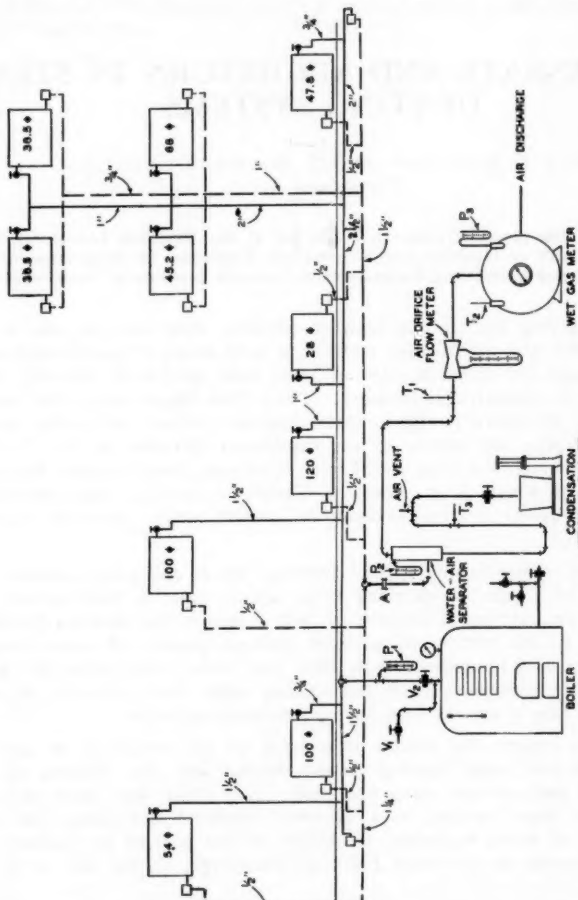

Fig. 1. Test Equipment and Heating System in the Residence Used in Making Gravity Study

and the size and location of the radiators are indicated. In this system steam was supplied by the boiler, the condensate was separated from the returning air near the point where the regular air relief valve had been located, and the

air was eliminated through the air orifice flow meter and the wet gas meter. The orifice in the orifice flow meter was made of a size approximating the orifice opening in air relief valves for heating systems of the sizes studied. As the Technical Advisory Committee on Pipe Sizes recommended orifices of 1/16 in. for systems containing up to 1,500 sq ft equivalent radiation, tests on the two systems were made with both 1/16 and 1/8-in. orifices. The wet gas meter used to measure the volume of air eliminated was sufficiently large so that the pressure drop through it never exceeded 0.1 in. of water column. Pressures were observed by manometer at the points P_1 to P_3 as indicated in the figure; temperatures were read by thermometer at points T_1 to T_3. The condensate was returned by gravity into the condensation weighing bucket.

Petroleum coke, which gave a very high and uniform rate of combustion, was used to fire the boiler in the residence. A steam pressure of 10 lb was developed in the boiler before the beginning of the test and this pressure was regulated by blowing the excess steam to the atmosphere through the relief valve, V_1. When conditions were ready for test, steam was turned into the system by opening and adjusting the valve, V_2, until the desired pressure on the supply side of the system was indicated by the manometer. The rates of returning condensate and air were observed each minute.

Fig. 2 is a diagrammatic sketch of the pipe system and the arrangement of the test equipment for vacuum pump operation as used for the tests in the 40-story Grant Building. The floors above the 33rd are heated by steam distributed through upfeed supply risers taken from the 10-in. supply main *BC*, which makes a loop above the false ceiling of the 33rd floor. In the sketch, the equivalent sq ft of radiation on each floor served by each pair of supply and return risers are indicated as a single unit, but actually this radiation in most instances included several units. The returning air and condensate from all return risers were collected through branches and mains between the 33rd and 34th floors at *A*, from which point they were carried to the condensate and air receiver and then to the vacuum pump, which delivered the air through the air orifice flow meter and the wet gas meter to the atmosphere, while the condensate was sent to the calibrated condensation measuring tanks. Pressures and temperatures were read on manometers and thermometers located at the respective points, P_1 to P_3 and T_1 to T_3.

In operating the system in the Grant Building by vacuum pump, the predetermined steam pressure was maintained constant in the supply main. Before starting a test the desired vacuum on the return side of the system, as indicated by the manometer, was obtained with the vacuum pump and through operation of valves V_1 and V_2. At the instant of starting the test the ten steam supply valves V_3 to V_{12} were opened; the time required in opening these valves never exceeded one minute. Except for the period immediately after steam had been turned into the system, when for a very few minutes the desired vacuums were not always maintained, the pressure in the return side of the system was held constant throughout a test. The weight of condensate and the volume of air removed from the system were observed at frequent intervals. The time when all radiators had become heated to steam temperature was taken as the end of the heating-up period, though the test was continued beyond this time to insure that a constant rate of return of condensate and air had been reached.

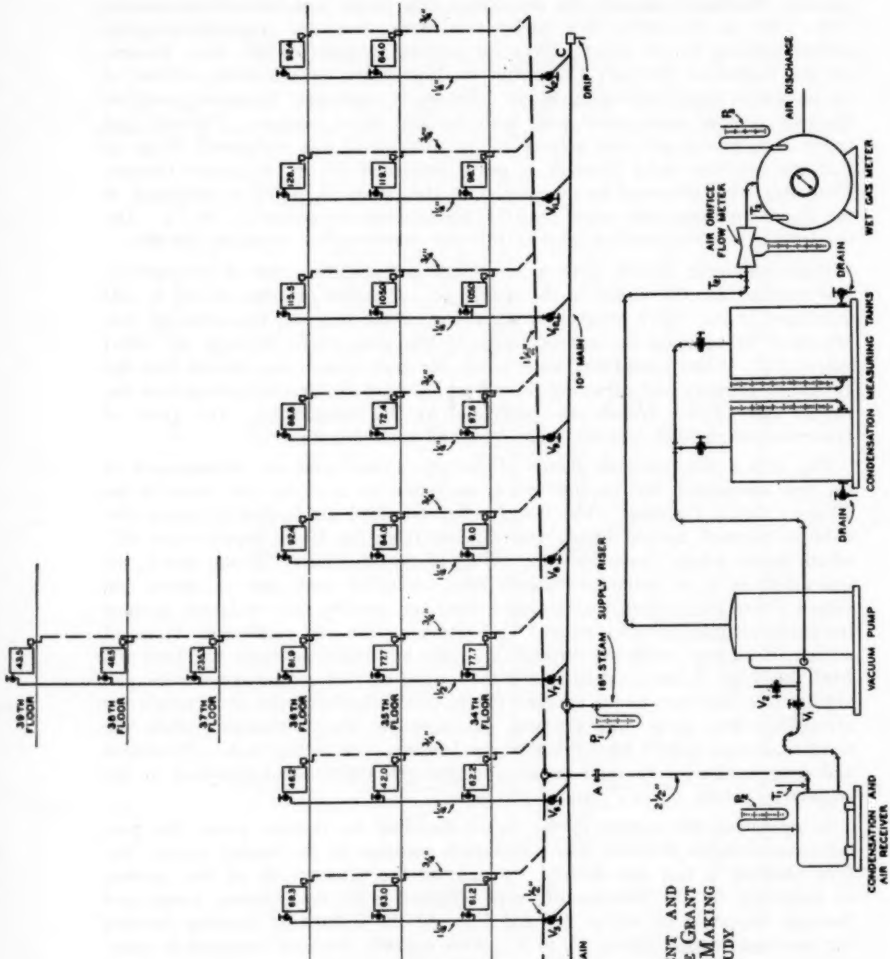

FIG. 2. TEST EQUIPMENT AND HEATING SYSTEM IN THE GRANT BUILDING USED IN MAKING VACUUM PUMP STUDY

Gravity return tests were conducted in the Grant Building in a manner similar to the vacuum pump tests, except that the gravity flow measuring equipment shown in Fig. 1 was attached at *A* instead of the vacuum pump arrangement of Fig. 2. The condensate and air were separated and measured in the same way as in the gravity test in the residence.

RESULTS

Three tests were made on the system in the Grant Building with gravity operation, six in the Grant Building with vacuum pump operation, and two tests were made on the gravity return system in the residence. Test data and conditions under which each test was made are tabulated in Table 1. In Fig. 3 the rates of returning condensate and air are plotted against elapsed time, from the instant when steam was turned into the system, for each of the nine

TABLE 1. TEST DATA AND CONDITIONS

Test Number	Test Location	Type of Test	Size of Orifice (In.)	Supply Pressure (Lb per Sq In. Absolute)	Return Pressure (Lb per Sq In. Absolute)	Pressure Difference (Lb per Sq In.)	Average Pressure in System (In. Hg Gage)	Steam Temperature (Deg Fahr)	Room Temperature (Deg Fahr)	Heating-up Period (Minutes)	Elapsed Time to Max Air Elimination (Minutes)	Elapsed Time to Max Condensate Elimination (Minutes)	Elapsed Time to Max Pressure Drop (Minutes)
1	Grant Bldg.	Gravity	1/16	16.27	14.24	2.03	+ 2.06	213.8	70	40	4.5	31.0	29.5
2	Grant Bldg.	Gravity	1/8	15.30	14.24	1.06	+ 1.08	212.2	78	37	0.5	26.0	15.0
3	Grant Bldg.	Gravity	1/8	16.27	14.24	2.03	+ 2.06	213.8	80	20	1.0	13.0	12.0
4	Grant Bldg.	Vacuum	None	13.91	12.83	1.08	− 1.90	207.4	80	20	1.5	9.5	7.5
5	Grant Bldg.	Vacuum	None	9.52	8.41	1.11	−10.88	188.0	80	27	0.5	19.0	19.0
6	Grant Bldg.	Vacuum	None	9.54	8.92	0.62	−10.39	189.5	80	15	2.0	7.0	7.0
7	Grant Bldg.	Vacuum	None	13.91	13.46	0.45	− 1.27	208.5	80	25	1.0	13.0	13.0
8	Grant Bldg.	Vacuum	None	9.54	8.77	0.77	−10.56	189.0	80	24	4.0	11.5	11.5
9	Grant Bldg.	Vacuum	None	11.97	11.34	0.63	− 5.43	200.5	81	21	3.0	11.0	11.0
10	Residence	Gravity	1/8	14.84	14.33	0.51	+ 0.53	211.6	78	33	2.0	8.0	7.5
11	Residence	Gravity	1/16	15.32	14.32	1.00	+ 1.02	212.4	76	50	2.5	5.0	5.0

tests in the Grant Building, and in Figs. 4 and 5 for the gravity tests made at the residence. Besides showing the curves for air and condensate, Figs. 4 and 5 graph the steam pressure maintained at the supply end of the system, and the pressure drop through the system from the steam supply to the end of the return just before the orifice. The separation between the steam supply pressure curve and the curve for the pressure drop through the system represents the pressure drop through the orifice.

It will be noted in these tests that immediately after steam was turned into the supply side of the system air began to be eliminated at the farthest end of the return. The rate of air elimination quickly reached a maximum, and then receded rapidly to a rate much lower, from which it continued to decrease more slowly until a constant rate of elimination was reached. The condensation, on the other hand, did not flow from the system until several minutes

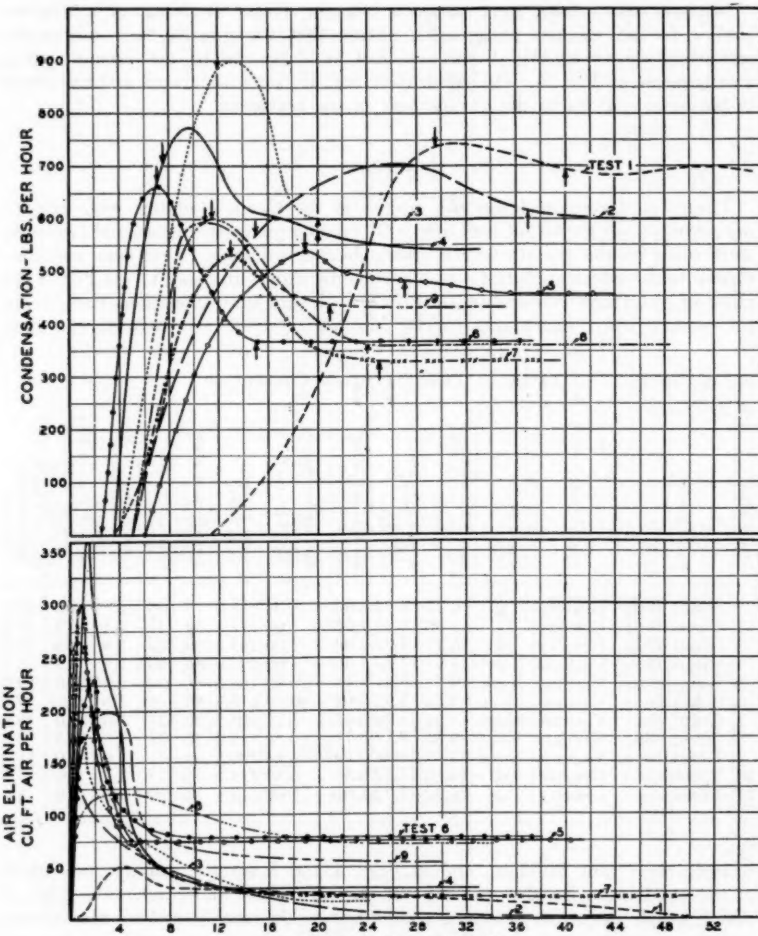

FIG. 3. RELATION OF ELAPSED TIME TO AIR ELIMINATION AND TO CONDENSATION RETURN IN THE HEATING SYSTEM AT THE GRANT BUILDING

after the steam was turned on. It reached its maximum rate slowly and then diminished to its constant rate. The arrows pointing up to the condensation curves indicate the end of the heating-up period.

In Fig. 3 the different curves for air and condensate show a marked similarity. Certain variations in the curves bear a definite relation to steam

pressure and pressure drop through the system; however, sufficient tests were not made to evaluate these relations accurately.

Conditions during steady heated-up operation at the Grant Building are shown in Fig. 6, where the lower curve gives the relation between the

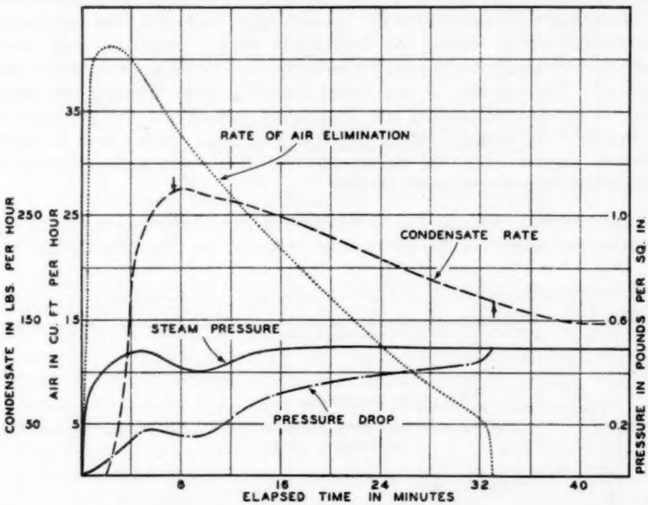

FIG. 4. RELATIONS BETWEEN ELAPSED TIME AND DATA FROM TEST 10 IN THE RESIDENCE—½ LB STEAM PRESSURE—⅛-IN. ORIFICE AIR RELIEF

observed rate of constant condensate return and the temperature difference between the steam in the system and the air in the rooms. The steam temperature was taken from steam tables for an absolute pressure equal to the mean between those of the supply and the return. As would be expected, the rate of condensation return increases with increase in temperature difference. The calculated curve shows the relation between the rate of condensate return and the temperature difference; it is based upon the installed radiation (2,510 sq ft) plus a ten per cent allowance for condensation in the return pipe, assuming 240 Btu emission per square foot for a temperature difference of 145 deg between steam and air, and assuming that for other temperature differences the condensation varies as the 1.3 power of the temperature difference. It will be seen that the rate of return in the Grant Building, as indicated by the observed curve, increases more rapidly with increased temperature difference than as indicated by the calculated curve.

Excepting for the small volume of non-condensable gases contained in the steam, the air elimination from a system should, if there is no leakage, decrease to zero upon complete heating-up. The constant rate of air elimination after heating-up therefore represents leakage which should be a function of the difference in pressure between the atmosphere and the inside of the system.

This rate of air leakage is plotted in Fig. 7 against pressure difference for the tests made under vacuum pump operation.

The relation between the maximum rate of condensate return during the heating-up period and the pressure drop through the system from the supply main to the end of the return is given in Fig. 8 for both gravity and vacuum pump tests in the Grant Building. As would be expected, the maximum rate of condensate return during the heating-up period increased with increased pressure drop through the system, or with the rate of advance of steam through the system. The relation in the Grant Building tests between the maximum rate of air elimination during the heating-up period and the pressure differences through the system during the times of these maximums is shown in Fig. 9, and indicates that the maximum rate of air elimination also increases with pressure drop through the system.

The most striking fact in Fig. 3 concerning the condensation return and the air elimination return is the short time required for the elimination of most

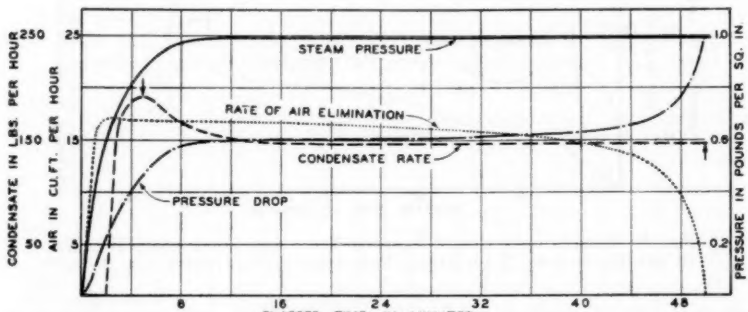

FIG. 5. RELATIONS BETWEEN ELAPSED TIME AND DATA FROM TEST 11 IN THE RESIDENCE—1 LB STEAM PRESSURE—1/16-IN. ORIFICE AIR RELIEF

of the air from the system. In all cases the maximum rate of air elimination comes much earlier than the maximum rate of condensation return. In fact, in all cases the air elimination rate has reached its maximum and receded to a much lower rate during the first five minutes; whereas the condensation return rarely approaches its maximum rate until a later time. This is a most significant fact bearing on the needed capacity of the return for carrying condensation and air during the heating-up period, for if the air elimination maximum were coincident with the condensation return maximum, the load on the system at this point would be greater and larger pipe would be required. The curves show that the tax on the capacity of the pipe caused by the air elimination comes earlier than and is rather independent of the tax on the pipe caused by the return condensate. These facts are shown in Table 2, in which comparative data are listed for analysis of the different tests studied.

Column 2 of Table 2 gives the total volume of air eliminated from the system

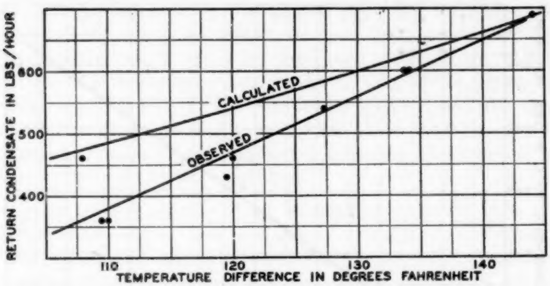

FIG. 6. OBSERVED AND CALCULATED RELATIONS BETWEEN TEMPERATURE DIFFERENCE (STEAM TO AIR) AND CONDENSATION RETURN

during the heating-up period, Column 3 gives the volume of air leakage into the system during the heating-up period because of the vacuum maintained, and Column 4 gives the difference between these two columns or the net air that would have been eliminated during the heating-up period had there been no in-leakage. It should be emphasized that, unless otherwise stated, volumes of air shown in the figures and tables are the volumes measured by the wet gas meter under the prevailing temperature and pressure in the meter, which were practically those of the atmosphere. Because atmospheric conditions were not had in the system before a test was started, its temperature being usually higher and its pressure usually lower than those registered on the meter, it is necessary to correct the metered volumes for the temperatures and pressures in the system to obtain the actual capacity of the system.

All of the air in the supply pipe and heating units should be eliminated during the heating-up period, and if the thermostatic traps work perfectly, steam should not enter the return. However, the hot condensate flowing through the return will so raise the temperature of the air contained therein

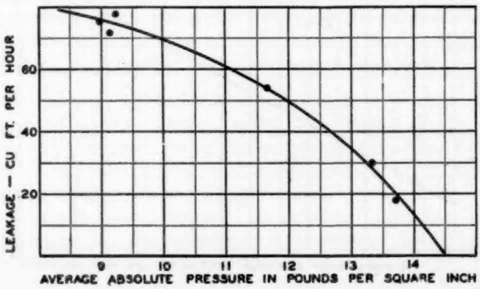

FIG. 7. RELATION BETWEEN AVERAGE ABSOLUTE PRESSURE AND AIR LEAKAGE

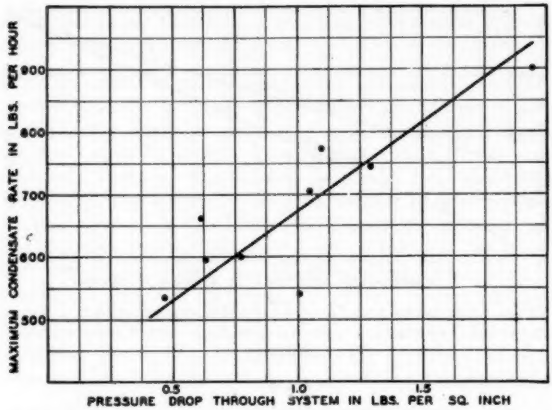

FIG. 8. RELATION BETWEEN PRESSURE DROP THROUGH THE SYSTEM AND MAXIMUM RATE OF CONDENSATION RETURN FOR ALL TESTS IN THE GRANT BUILDING

and expand its volume that a large part of it will also be forced out. The values given in Column 4 for the net volume of air eliminated from the system during the heating-up period are corrected in Column 5 for the prevailing presure and the estimated temperature within the system.

Values shown in Column 5 should therefore represent the volume of the air contained in the system before the steam was turned on, or the volumetric capacity of the supply pipe, the radiators, and a part of the return piping.

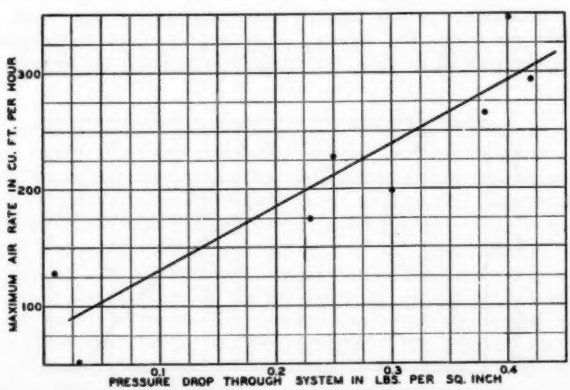

FIG. 9. RELATION BETWEEN PRESSURE DROP THROUGH THE SYSTEM AND MAXIMUM RATE OF AIR ELIMINATION FOR ALL TESTS IN THE GRANT BUILDING

A calculation of the volumes of the supply pipe, the heating units, and the returns, based on the size and length of pipe and the size and volumetric capacity of the heating units, gives for the Grant Building the values 10.5, 4.4, and 3.5 cu ft for the supply pipe, heating units, and return pipes, respectively, or a total volume of 18.4 cu ft for the entire system; and for the residence, 3.6, 11.1, and 0.8 cu ft, respectively, or a total of 15.5 cu ft. It will be noted that the corrected volumes of air eliminated from the two systems check fairly well with their total volumetric capacities.

Column 6 gives the net volume of air eliminated up to the time of maximum rate of condensation return, and Column 7 reduces this volume to a percentage of the volume of air eliminated during the entire heating-up period. Likewise, Column 8 gives the net volume of air eliminated up to the time when the rate of condensation is half of its maximum, while Column 9 reduces this volume to its percentage of the volume eliminated during the entire heating-up period. It is of interest to note that for the tests in the Grant Building the air eliminated from the system up to the time of maximum condensation represents a range of from 71 to 100 per cent of the total eliminated during the heating-up period, an average for all tests of 92 per cent, while for the period up to the half-maximum point the air eliminated represents an average of 75 per cent of the total eliminated for the entire heating-up period. The similar percentage relations are lower in the tests made in the residence. This can be explained by reference to the curves for these tests in Figs. 4 and 5. Although the pressure in the boiler was built up to ten pounds before the steam was turned into the system, the small steam storage capacity of the boiler, compared with the larger volume of the supply piping, resulted in an immediate drop in steam pressure, which did not again reach the desired value until several minutes later. Also, during the early part of the tests, the greater part of the pressure difference between the steam supply and the atmosphere was required to force the air through the orifice, so the rate of air elimination did not quickly reach its peak, which was broader and continued for a longer time. The 1/16-in. orifice used in Test 11 in the residence and in Test 1 at the Grant Building was obviously too small to allow the air to leave the systems quickly. The retardation of the rate of air elimination from the system by the small orifices should not necessarily be considered an undesirable limitation, for one of the accepted, although not generally recognized, functions of a good air relief valve is to control the rate of air elimination and therefore the rate of steam supply to the system during the heating-up period to avoid noisy operation during this time.

Column 10 of Table 2 gives the ratio of the rate of total air elimination to the rate of condensation return when this return is at its maximum. Columns 11, 12, and 13, give, respectively, the condensation rate after it had become constant at the end of the heating-up period, the maximum condensation rate during the heating-up period, and the percentage which this maximum rate is of the constant rate. It will be observed that the ratio of air to condensate at the time of the maximum condensation load ranges from less than 0.01 cu ft per pound of condensate to 0.16 cu ft, giving an average for all tests in the Grant Building of 0.08. These ratios for tests in the Grant Building are plotted against the average absolute pressures within the system in Fig. 10. It will be observed that the ratio increases as the absolute pres-

TABLE 2. ANALYSIS OF TEST DATA

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Test Number	Total Air Eliminated Heating-up Period (Cu Ft)	Total Air Leakage Heating-up Period (Cu Ft)	Net Air Eliminated Heating-up Period (Col 3−Col 2) (Cu Ft)	Corrected Net Air Eliminated Heating-up Period[a] (Cu Ft)	Net Air Eliminated to Time of Max Cond (Cu Ft)	Percentage $\frac{\text{Col. 6}}{\text{Col. 4}} \times 100$	Net Air Eliminated to Time of Half Max Cond (Cu Ft)	Percentage $\frac{\text{Col. 8}}{\text{Col. 4}} \times 100$	Ratio—Air to Cond. at Time of Max Cond (Cu Ft per Lb)	Constant Rate Cond Return (Lb per Hour)	Max Rate Cond Return (Lb per Hour)	Percentage $\frac{\text{Col. 12}}{\text{Col. 11}} \times 100$	Max Pressure Drop 1¼-In. Pipe (Oz per 100 Ft)	Percentage that Max Cond Rate is of Designed Cond Rate	Ratio—Air at Max Cond to 143% of Designed Load (Cu Ft per Lb)
1	15.4	0.0	15.4	15.6	14.1	92	9.9	64	0.02	690	744	108	0.45	118	0.02
2	16.8	0.0	16.8	16.9	14.6	87	10.8	64	0.01	600	705	118	0.22	112	0.01
3	17.9	0.0	17.9	18.0	16.9	95	9.2	51	0.04	600	900	150	1.20	143	0.04
4	28.5	10.0	18.5	20.8	17.2	93	15.2	82	0.04	540	771	143	0.77	122	0.04
5	39.6	33.0	6.6	11.3	6.5	100	6.5	100	0.14	462	540	117	0.46	86	0.08
6	26.0	19.3	6.7	19.3	6.7	99	5.7	84	0.13	360	660	183	0.96	104	0.09
7	24.4	9.3	15.1	16.3	14.3	94	14.1	93	0.05	330	534	162	0.20	83	0.03
8	38.6	28.8	9.8	16.2	7.0	71	4.8	49	0.16	360	600	167	0.84	95	0.01
9	32.2	18.9	13.3	16.9	12.4	93	11.2	84	0.11	432	594	138	0.55	94	0.01
Averages				16.8		92		75	0.08			143			
10	11.8	0.0	11.8	11.8	8.3	70	2.3	18	0.14	144	155	108	0.02	97	0.10
11	12.5	0.0	12.5	12.5	1.2	10	0.5	4	0.09	147	194	132	0.00	121	0.07

[a] Volume in Column 5 corrected for temperature and pressure in pipe.

sure in the system decreases, or as the vacuum applied increases. This increase in the ratio with vacuum applied is obviously the result of air leakage into the system. The ratios due to elimination of air, after correction for leakage, range down from 0.05 cu ft per pound as shown by the points *X* at the lower part of the figure. The tests in the residence, where there was no in-leakage of air, show ratios greater than 0.05 cu ft per pound of condensate at the time

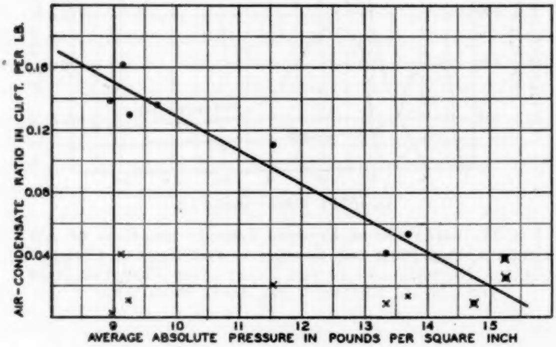

FIG. 10. CURVE SHOWING RELATION BETWEEN THE AVERAGE ABSOLUTE PRESSURE AND THE RATIO OF TOTAL AIR TO CONDENSATE AT THE TIME OF MAXIMUM CONDENSATION; POINTS "X" SHOW RELATION BETWEEN THE AVERAGE ABSOLUTE PRESSURE AND THE RATIO OF NET AIR TO CONDENSATE AT THE TIME OF MAXIMUM CONDENSATION FOR ALL TESTS IN THE GRANT BUILDING

of maximum condensate return, due to the slowing up of elimination of air from the system because of the orifices used. This higher ratio is therefore accompanied by decreased maximum condensation rates, as is indicated in Column 13, which tabulates the percentage relation between the maximum and constant condensation return rates. In the Grant Building the maximum rate of condensation return ranged up to 183 per cent of the constant condensation rate, with an average of 143 per cent, while in the residence the maximum condensation rates are 108 and 132 per cent of the constant rate for the two tests. This indicates that the maximum rate of condensation return is decreased during the heating-up period, when because of the retardation of elimination of air at the time of maximum condensation, the ratio of air to condensate is increased. As the ratio of air to condensate during the heating-up period increases, tending to indicate a greater load on the system, the condensation rate decreases and compensates the need for larger sized pipe. In general, those tests in which during the heating-up period the maximum condensation rate is the greatest percentage of the constant condensation rate are tests with a high vacuum or low absolute pressure, resulting in both a lower maximum and a lower constant condensation rate, which compensation again tends to lessen the need for excessive pipe sizes.

It is of interest to study the combined tax on the return piping resulting

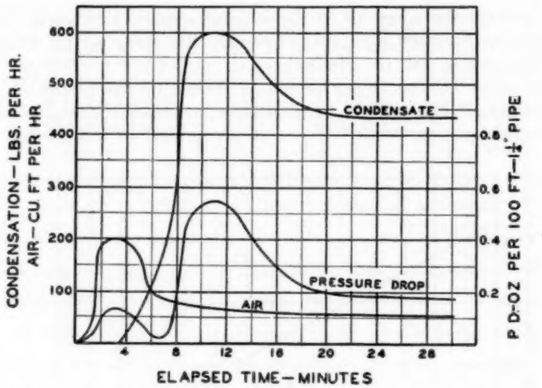

FIG. 11. RELATION OF ELAPSED TIME TO THE RATE OF AIR ELIMINATION FROM THE SYSTEM TO THE RATE OF CONDENSATION RETURN, AND TO THE CALCULATED PRESSURE DROP FOR 1¼-IN. RETURN. ALL DATA FROM TEST 9

from the various combinations of water and air flowing during and after the heating-up period. Results of Test 9 at the Grant Building are shown in Fig. 11, in which the air elimination and condensation return curves are plotted together with a derived curve showing pressure drop for 1¼-in. pipe. The pressure drop during the time when air only was flowing through the return was calculated from an accepted formula [2] for the flow of air in pipe. For both air and water flowing through the pipe the pressure drop was calculated from the following formula developed in a previous Laboratory paper, p. 179:

$$L = (2.18\ P^{0.04520} - P^{0.30808} \log R)\ (157.08\ d^2 + 80) - 500 \qquad (1)$$

where

L = condensation load in pounds per hour
P = pressure drop in ounces per 100 ft
R = ratio—cubic feet of air per pound of condensate
d = actual internal diameter of pipe in inches.

It will be noted from the curve that the pressure drop, resulting from carrying only air at the high elimination rate immediately after turning on the steam and before the condensation return becomes an important factor, is small and increases to a maximum at the time of the maximum condensation rate, from which it recedes as the rate of air elimination decreases.

Column 14 of Table 2 lists for each test the pressure drop in ounces per 100 ft of 1¼-in. pipe as calculated for the point of greatest pressure drop, indicated by the arrows pointing downward to the curves, Figs. 3 to 5. In the few cases where the maximum pressure drop did not coincide with the point of

[2] *Marks Handbook*, Third Edition, p. 376.

maximum rate of condensate return, the calculation of the pressure drops for the two points showed the variation to be negligible. Since the maximum pressure drop or the maximum tax on the system comes usually at the time of the maximum rate of condensation return, it depends upon the maximum rate of condensation return and the related rate of air elimination.

It is not desirable to take into account all the varying factors including, among others, absolute pressure, pressure drop, orifice size, and volumetric capacity of the system in determining pipe sizes for a given system. However, it would seem expedient to base pipe sizes on a condensation return rate equal to an accepted percentage of the condensation rate for the installed radiation under standard conditions with a logical ratio of air to condensate.

It might appear that a system operating under a high vacuum would require larger pipe sizes than one operating at a high absolute pressure, because with increased vacuum on the system there was an increase in the percentage relation which the maximum condensate return rate bore to the constant condensate return rate and also in the ratio of air to condensate at the time of maximum condensate. This is, however, because the values on which the percentages and ratios are based are smaller for the vacuum system. In order to bring this fact out, the percentage which the maximum condensation return is of the steady condensation return is given in Column 15, as calculated from the installed radiation under standard conditions. In the Grant Building this designed condensation load is $\frac{2510}{4} = 628$ lb per hour, and in the residence it is $\frac{640}{4} = 160$ lb per hour. Column 16 gives the ratio of the air elimination at the time of maximum condensation return to the maximum rate of condensation return, which is based on the largest percentage found in Column 15, or 143 per cent, and on the calculated constant condensation rate for the installed radiation operating under standard conditions. For the Grant Building this rate of condensation return is 143 per cent of 628, or 900 lb per hour, and for the residence it is the same percentage of 160, or 229 lb per hour.

It is of interest to speculate on what the increased tax on the system in the Grant Building would have been if tubular cast iron heating units had been used instead of copper radiation. The total volumetric capacity of supply, return and radiation in the Grant Building was found to be 18.4 cu ft, made up of 14 cu ft in the piping and 4.4 cu ft in the heating units. The volumetric capacity of this type of copper radiation was found by measuring twelve heating units of different size which averaged 0.00174 cu ft per square foot of radiation. In an earlier Laboratory study,[3] measurement of fifteen tubular cast iron radiators of various sizes and four different makes showed an average volumetric capacity of 0.0142 cu ft per square foot of radiation. Therefore, if the same number of equivalent sq ft of tubular radiation had been used in the Grant Building, the volumetric capacity of the radiation would be 35.7 cu ft, which, increased by piping, would give 49.6 cu ft as the total volumetric capacity of the system—a total volumetric capacity about three times as great as was found with copper heating units. It would be difficult

[3] Capacity of Dry Return Mains for Steam and Vapor Heating Systems, by F. C. Houghten and Carl Gutberlet, A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 481.

to determine accurately the effect which this increased volume of air would have on the tax on the system during the heating-up period without testing a number of similar systems containing different types of radiation. The assumption may be made, however, that the rate of progress of water through the system would be slowed up, allowing a large part of the excess air to be expelled before the condensation load reached its maximum, and that the increased ratio of air to condensate at the time of maximum condensation return would not be proportionately larger. Instead of air ratios of 0.05 cu ft per pound of condensation or less at time of maximum condensation return as indicated in Fig. 10 for a system operating at atmospheric pressure, and therefore without leakage, the ratio of air to condensate under the assumed conditions using tubular radiators may not be increased beyond 0.1. However, it should be emphasized that this assumption is not based on test data.

The increased rate of air return during the time of maximum condensation return for a system operating under vacuum as shown in Fig. 10 should not indicate the need of larger returns for vacuum pump systems designed to operate at considerable vacuum, for the reason that such systems should be designed to operate at high vacuum only during mild weather when the demand on the system is below maximum. It would be expected that a vacuum pump system should operate at or above atmospheric pressure at the time of maximum heating demand, thus making its maximum demand during the heating-up period no different from that of a gravity system operating at atmospheric or slightly higher pressure.

CONCLUSIONS

The foregoing facts should be considered, together with the fact that the maximum rate of condensation never reached more than 143 per cent of the designed condensation load, and the fact that in the system with copper radiation the air-condensation ratio at the time of maximum load never exceeded 0.05 cu ft of air per pound of condensate when the system operated at or above atmospheric pressure, or when there was no leakage. From such consideration it would seem reasonable to recommend that return pipe sizes for vacuum systems be based upon a maximum condensation load during the heating-up period of 150 per cent of the designed condensation load under standard conditions, and a ratio of 0.10 cu ft of air per pound of condensate, allowing possibly a slightly smaller ratio of air for systems containing radiation with small volumetric capacity and a slightly higher ratio for systems containing radiation with large volumetric capacity. Return pipe sizes in gravity systems should allow a larger factor of safety, and hence it is recommended that they be based on 175 per cent of the designed condensation load under standard conditions and a ratio of 0.15 cu ft of air per pound of condensate.

No. 955

SEMI-ANNUAL MEETING, 1933

WITH the best attendance in four years, the Society held its Semi-Annual Meeting at the Hotel Statler, Detroit, June 22-24, approved the submission of two codes to the membership and discussed a series of interesting papers during the three technical sessions. A well balanced program was provided and the entertainment planned and carried out by the Michigan Chapter members satisfied every taste.

At the meeting of the Council on Wednesday, June 21, the nomination of five members to serve for a three-year term on the Committee on Research was made as required by the By-Laws of the Society. Dates were selected for the 40th Annual Meeting, February 5-8, 1934, with the New York Chapter as hosts and it was the decision of the Council that the Semi-Annual Meeting 1934 will be held under the auspices of the Philadelphia Chapter. Revised rates for GUIDE advertising in the 1934 edition were approved and resolutions of appreciation were adopted for the cooperation given by the *American Oil Burner Association* and the *American Society for Testing Materials.*

A joint meeting of the Membership Committee and the Chapter Relations Committee was held on June 22. Chapter relations work was discussed and it was planned to send detailed suggestions to the chairman of the Program Committee of each Chapter. The importance of membership work was emphasized and a definite program was outlined.

The first session of the Semi-Annual Meeting 1933 convened at the Hotel Statler, Detroit, Mich., on Thursday, June 23, at 9:30 a. m.

President Jones announced that the Society was now functioning under the new Constitution, By-Laws and Rules, adopted by letter ballot of the members April 3, 1933.

Report of Committee on Atmospheric Dust and Air Cleaning Devices

H. C. Murphy, Chairman of the Committee on Atmospheric Dust and Air Cleaning Devices, was introduced by President Jones and presented the preliminary draft of the A. S. H. V. E. Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work[1] with the following Foreword:

The resolution to formulate a Code for Testing Air Cleaners was introduced at the Annual Meeting of the Society at Buffalo, N. Y., in January, 1926. The need for some standard method of testing and rating air cleaning devices was even then apparent. Rapid developments in the field of air conditioning have made such a code even more essential today.

Investigations were first directed to dust measuring instruments. Most of this

[1] The A. S. H. V. E. Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work, which was adopted as amended, appears on p. 225.

work was done at the University of Minnesota by Prof. F. B. Rowley and John Beal and published in TRANSACTIONS of the Society, Vol. 33 and 34, 1928 and 1929.

It became apparent that a code suitable for laboratory testing of air cleaning devices used in general ventilation work would be entirely unsuited for rating cyclones, dust separators and the like or for the selection and rating of devices intended to supply air for dust hazardous occupations in accordance with the standards of the U. S. Public Health Service.

It therefore seemed advisable to limit the use of this Code as defined under *Scope* and it was also decided to recommend for this particular Code the use of the weight method and non-proprietary instruments for determining cleaning efficiencies, substantially in accordance with the procedures used by the U. S. Bureau of Standards, U. S. Department of Agriculture, by steel mills and other industrials having large numbers of determinations to make.

The present arrangement of the Code is based on the procedure developed in air cleaner testing by Mr. Stig Sylvan. The Committee is also indebted to Mr. Sylvan for the design of much of the apparatus used.

Mr. Samuel R. Lewis, consulting engineer, has in the course of approximately 100 tests of various air cleaners, developed some valuable simplifications of the Code which have been incorporated in the present draft.

In the cooperative tests conducted at the University of Minnesota for the National Warm Air Heating Association, the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and various air cleaner manufacturers, Prof. F. B. Rowley has instituted some further modifications which have seemed desirable to incorporate in the present draft of the Code.

COMMITTEE ON ATMOSPHERIC DUST AND AIR CLEANING DEVICES

H. C. MURPHY, *Chairman*

J. J. BLOOMFIELD	DR. E. V. HILL	F. B. ROWLEY
ALBERT BUENGER	S. R. LEWIS	GAMES SLAYTER
PHILIP DRINKER	H. B. MELLER	DR. S. W. WYNNE

Discussion of the Report

FRANK THORNTON, JR. (WRITTEN): The Proposed Code is in general quite satisfactory, although actual use will undoubtedly lead to further developments and refinements.

There is one point in particular about which it is felt that more complete details should be given in the Code. This is the definition of Standard Dust in Paragraph B-9. The words, powdered lamp black, might cover almost any quality of lamp black. Some kinds of lamp black are very finely divided and when thrown into the air will float about in the air for a long period of time. Other grades of lamp black when treated in the same way will very quickly settle to the floor. Also, the ashes, which will pass a screen of 200 mesh may be largely composed of particles which will just pass such a screen, or on the other hand they may be largely composed of particles which are very much finer.

I am unable to offer constructive suggestions regarding this definition, and therefore can only point out that it might be unwise to formally adopt a Code of this kind containing such an important specification as this definition for Standard Dust, unless it is definitely stated that work will be continued and that the Code will be subject to further revision next year.

DR. LEONARD GREENBURG: The problems which involve the sampling and determination of atmospheric dust are among the most difficult in the field of ventilation and industrial hygiene. Mr. Murphy, the Chairman of your Committee, and his colleagues have, by painstaking effort, made real contributions to the problem of testing air cleaning devices. It may be that some of the members of the Society disagree with portions of the Code which has been submitted for your consideration this morning. I am cognizant of the fact that changes will be required in this Code from time to time, but I feel that these are minor in comparison with the strides

made by your Committee, and I urge the adoption of this Code by the membership as a whole.

After further discussion on the subject, it was moved and seconded that the Code prepared by the Committee be adopted as amended and submitted to the membership for approval by letter ballot.

On motion of Mr. Murphy, seconded by J. H. Hale, it was

Voted, that the Committee's report be accepted and that the Code be adopted in its present form with the understanding that changes noted and briefly mentioned will be included and the Committee be continued for further consideration of similar subjects.

On motion of Mr. Murphy, seconded by Professor Rowley, it was

Voted, that a Committee be appointed to cooperate with other interested agencies in endorsing the use of public funds to the extent provided in Section 202 of the National Industrial Recovery Act as outlined above.

R. H. Carpenter presented a resolution from the New York Chapter relating to the proposed program of business revival and suggested the appointment of a Society committee to cooperate with other organizations of engineers, architects and others in furthering the reconstruction program. These ideas were presented in the form of resolutions prepared by the committee of the New York Chapter consisting of Perry West, A. F. Hinrichsen and S. A. S. Patorno.

On motion of Mr. Carpenter, seconded by Mr. Howatt, it was

Voted, that these resolutions be referred to the Advisory Council.

The motion was unanimously adopted.

W. W. Timmis reported on the progress of the work of the Committee on Corrosion of the *New York Real Estate Board* with which the Society is cooperating.

A message of greeting from the *American Oil Burner Association* was received from Pres. Morgan J. Hammers.

Report of Committee on Code for Testing and Rating Convectors

At the second session, Friday, June 23, 9:30 a. m., President Jones presented R. N. Trane, Chairman of the Committee on Code for Testing and Rating Convectors, who presented the Committee's report which covered the Hot Water Section of the A. S. H. V. E. Standard Code for Testing and Rating Concealed Gravity Type Radiation.[2]

F. D. MENSING (WRITTEN): I have gone over the A. S. H. V. E. Standard Code for Testing and Rating Concealed Gravity Type Radiation, and note under *A. Purpose,* that, "The object of this Code is to provide a practical method for testing and rating convectors which may be readily used by manufacturers, and which will provide the necessary information for the heating engineer in the design of the heating plant," and that, "This code describes a boiler load method of rating." Further, that, "The term convector as used in this code shall apply to any type of radiator installed within a wall, casing, or enclosure, having an inlet and outlet for the circulation by gravity of the air in the room to be heated."

Inasmuch as this is to be a practical method of testing which may be used by

[2] The A. S. H. V. E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Hot Water Code), which was adopted as amended, appears on p. 237.

manufacturers to provide information for heating engineers in the designing of a plant, and inasmuch as the definition of convector will take in any type of enclosed radiator, ferrous, non-ferrous, fin type or prim surface type, when the same is enclosed; it is quite obvious that any method of testing must be one that will provide conditions similar to that found on the average hot water job, installed according to the present knowledge of the art.

A perusal of this paper shows that the water is to be voided through the convector by a gravity head, and is to discharge into an open tank. The static head produced is not the function and the location of the radiator itself and design of the piping system, but is an arbitrary assumption. Now it so happens that in all hot water jobs, the circulation in the radiator is produced by the difference in head in the flow and return line of the radiator, due to the cooling effect of the radiator. In other words, the energy required to circulate the water is produced within the system itself. To visualize how small these static heads are, the writer will give records taken in his own home, when outside temperature was 45 deg, which is about the average winter temperature in Philadelphia.

Temperature of water returning to the boiler	101 deg.
Temperature of water leaving the boiler	117 deg.
First floor radiator entering	116 deg.
First floor radiator leaving	106 deg.
Riser supplying 100 ft radiation in 4 radiators, 2 on second floor and 2 on third floor, temperatures taken at first floor level—flow	111 deg.
return	104 deg.

By comparing these temperatures, it will be noted that there is 16 deg rise in the boiler, 11 deg drop on the first floor radiator, 7 deg drop in riser supply, 4 radiators total a little over 100 ft, two on second floor and two on third floor; distance from ceilings being approximately 10 ft.

It is quite obvious that the difference in head due to the temperature drop is small compared with temperature drops that are likely to be assumed. Further, what temperature drop would be used in the case of this job when the temperature drop in the riser supplying two floors and four radiators is less than the drop in the case of the first floor radiator.

It is a fact well known to heating engineers and contractors familiar with hot water installations, whether gravity or forced, that each radiator has its own pump. They proportion the lines, take connections from the main and to the return and locate the return so that the radiator may so far as possible handle the same amount of water per square foot of surface.

When a radiator does not properly heat, the common term is that *it does not circulate;* that is, the radiator does not circulate, showing that the trade well recognizes that it is the radiator that circulates.

It is obvious that any test made with the proposed method based on assumptions unsupported by conditions not found on the job will produce results that are useless to the heating engineer or contractor laying out a job.

In the data sheet shown on page 242, provision is made for the stack height of the convector. The Code recognizes that the convector should handle the air through its structure. This being so, is it not just as logical that the convector should as a part of its duty handle the amount of water that it requires? In other words, if the Code is justified in using an assumed flow of water through the convector, is it not just as scientific to make an assumption on the amount of air handled by the convector?

In THE GUIDE, information is given on the method of laying out hot water systems, which information is generally satisfactory. Should the Society adopt this Code and should any one attempt to install a mixed job of ferrous radiation and convectors on the same system, using the methods as laid down in the Code, the writer knows by his experience and others', as does every member of the Society who has had experience, that the job so laid out would be a failure.

E. G. SMITH (WRITTEN): Paragraph 22 specifies that the water temperature drop

shall be 20 deg, presumably 20 F. This seems to be too low because, except in some one-pipe systems, radiators practically never run at temperature differentials below 20 F. Forced circulation systems are usually run at either 20 F or 30 F. The standard pipe sizes recently adopted by the *Heating and Piping Contractors National Association* call for temperature differentials ranging from 37½ F to 22½ F with an average of 31. Thirty degrees Fahrenheit is therefore much nearer to the average found in practice than the proposed 20 deg.

Paragraph 21 specifies that the inlet water temperatures shall be varied from 220 F to 100 F. The designer of a heating system is interested in the output of the radiators when the system is working at or near the maximum capacity for which it is designed. The inlet radiator water temperature for which a system is designed is almost never lower than 180 F. Consequently, it would seem to be a waste of time to test radiators at inlet temperatures much below 180 deg. To do so is the practical equivalent of testing a steam radiator when it is only partly filled with steam.

In view of the above, the writer would like to suggest the following changes:

The first sentence of Paragraph 21 to read: The convectors shall be tested with not less than four different inlet water temperatures ranging from 170 F to 230 F.

Paragraph 22 to read: The convector shall be tested with a water temperature drop of 30 F, with an allowable variation of ±1 F.

In Paragraph 28, the following to be inserted just below the formula: Results are to be checked by plotting the outputs under standard conditions for all lengths of heating element on a single set of coordinate axes, and if the resulting curves are not all the same shape, some or all of the tests are to be repeated until a set of similar curves is obtained.

The sentence which will follow the above insertion to be changed to read: Tables should show average water temperatures of 215, 195, 175, and 155 F for each width of heater for various lengths and heights, with the resulting Btu outputs and the pressure drops corresponding to a 30 F drop in water temperature, as suggested in the sample table shown in Fig. 3.

In Fig. 3, line 6: change 180 to 190.

In Fig. 3, last line: change $\frac{H}{20}$ to $\frac{H}{30}$.

In Fig. 3, just above the table insert: Average water temperature 175 F.

The reason for suggesting that the average water temperature be given more prominence than the inlet water temperature is because when the average water temperature is known, the heat output is practically independent of the water temperature drop in the convector. For this reason any designer using a temperature drop other than that used in the test, finds it very convenient to have the average water temperature given.

F. W. HVOSLEF (WRITTEN): The A.S.H.V.E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Hot Water Code) should set up operating conditions to simulate as closely as possible those which will be encountered in the field. The code as submitted does not do this and it is my opinion that data obtained with the method outlined will be absolutely inaccurate and therefore valueless. I offer a few detailed comments and recommendations and also a brief general discussion of the problem.

Paragraphs 3-15: To this section I believe should be added a sentence as follows: Modifications in apparatus and technique which do not materially affect the final result shall be permissible.

I make this recommendation because in our laboratory we use an apparatus which differs in many respects from that outlined and we feel that we get very accurate results from this equipment.

Paragraph 17: The operating air temperature range of 60 deg to 75 deg is unnecessarily wide and not conducive to accuracy in results. The minimum temperature can easily be established at 65 deg because it is always possible to raise a lower tem-

perature by the use of an auxiliary radiator. I believe that the maximum temperature should preferably be held at 72 deg in the interest of accurate results.

Paragraph 20: Because of the fact that we use a longer test period than that contemplated in this code we feel that this paragraph should be revised as follows: The convector shall be vented before starting the test and during the test as necessary . . .

We make this recommendation because if a longer test period is used we have found that there is some accumulation of air from the water passing through the convector.

Paragraph 22: This paragraph is the worst feature of the proposed code. It is set up on the assumption that water temperature drop in a radiator can be controlled in practice, whereas actually the temperature drop is a result of surrounding conditions. The heat output and temperature drop in a convector is affected by the following:

1. Size of convector
2. Height of stack
3. Entering air temperature
4. Entering water temperature
5. Water friction through the convector
6. Friction in the piping

It is obvious that the temperature drop through a long convector will be greater than that through a short convector and the differences may be very important.

Inasmuch as the heat output of a convector is proportional to the stack height it is also obvious that the temperature drop will be proportional to the stack height.

The temperature drop through the convector will be materially affected by the temperature of the air which is an important reason for holding the entering air temperature to a closer limit than contemplated.

In any given convector if the temperature drop is 20 deg with 200 deg entering water temperature the temperature drop cannot possibly be 20 deg when the water temperature is greater or less than 200 deg. It is obvious that the temperature drop would start at 0 as the air temperature and water temperature coincide and would increase in some geometric proportion as the water temperature increases.

The temperature drop through the convector is proportional to the friction within the convector and in many convectors, of cast iron construction, the friction is so low as to be almost unmeasurable. Consequently it is obvious that a 20 deg temperature drop might be reasonable for certain types of small tubular convectors under certain conditions and entirely unreasonable for convectors of other construction.

The temperature drop through the convector may be very vitally affected by the size of the connecting piping. It is obvious that the size of the connecting piping may be of minor importance with small tubular convectors where the convector friction is high; on the other hand where the convector friction is very low, as in the case of cast iron convectors, the size of connecting piping is of vital importance. We have found in our laboratory that the temperature drop through a given convector under fixed conditions may vary over 100 per cent by changing the supply and return piping one or two sizes.

Paragraph 25: The wording of this paragraph should be more specific and in my opinion should set up a minimum test period. With the very brief test period contemplated in the code the result of error is greatly magnified.

The code sets up means of measuring friction through the convector and tabulating the readings. As I have previously stated, in certain types of convectors the friction loss is so very low that the readings are almost microscopic. In any case friction readings are of questionable value because of their doubtful accuracy and because the average heating man has no conception of their meaning.

It is not my purpose to rewrite this code, I merely offer what I consider to be constructive criticism and I wish to explain briefly some of the things that we have done in an effort to arrive at accurate data concerning hot water performance of convectors.

We have installed convectors on a typical heating system on first, second and third floor levels.

On each level we have installed convectors of different sizes and types and we have provided them with stacks of various heights.

We have connected these convectors to the mains with branches and risers of various sizes in order to determine the optimum.

With room temperature conditions as nearly 70 deg as possible, we have determined the temperature drop through the convector under various conditions mentioned.

The various convectors have then been installed in the test room and the rate of water flow through the convector has been adjusted to give the same temperature as was found on the typical heating system. Under these conditions the performances have been measured.

Possibly your committee may find some food for thought in these comments. I realize that I propose to vastly complicate your problem, but I do not believe that the committee's solution of the problem is acceptable.

M. G. STEELE: I had not noted until reading the Code over this morning that in Paragraph 29, and Fig. 3, the use of the so-called *stack height* is specified for describing the effective height of the convector unit as a whole.

It has for some time been the practice of several leading manufacturers to use instead the *enclosure height,* which is the distance from the floor level to the roof, or top of the enclosure. It has been found by some that the term, *enclosure height,* is more definite and makes less confusion.

For instance, in determining the height of the necessary wall recess for a given convector unit it is merely necessary to allow a small clearance (for example ¼ in. or ⅜ in.) over the *enclosure height* (which is in effect the height of the metal box to be inserted in that recess). And again, capacity ratings based on *enclosure height* with definite standards of type and location of grilles, free air opening of same and height of convector from the floor, provide an accurate measure of capacity and performance.

I feel that the Code as it stands should be amended to include the use of *enclosure height* at least as an alternate to *stack height.*

PROF. A. P. KRATZ: Graduated control of the electrical heater can be obtained by employing a motor driven rheostat. This apparatus is rather expensive, however, and uniform control can be accomplished by means of the thermostat and contactor if two parallel circuits are employed. One circuit, not connected through the contactor, is adjusted manually by a rheostat so that it is not quite sufficient to maintain the required water temperature. The control is then effected by the second circuit connected through the contactor operated from the thermostat. The current in this circuit is made a very small proportion of the total current so that large temperature over runs do not occur when the contactor is closed.

At the conclusion of the discussion, on motion of Prof. F. E. Giesecke, seconded by J. D. Cassell, it was

Voted that the report of the Committee be accepted as amended and submitted to the Society for adoption by letter ballot. Also, that the Committee be continued.

This motion was carried unanimously.

Pres. W. T. Jones announced that ten past presidents of the Society were in attendance and he expressed his gratification on being able to introduce Dean F. Paul Anderson, W. H. Driscoll, John F. Hale, L. A. Harding, E. Vernon Hill, S. R. Lewis, J. I. Lyle, J. R. McColl, F. B. Rowley and F. R. Still.

President Jones introduced P. M. O'Connell, President of the Pacific Northwest Chapter, who held the long distance record for travel to the meeting.

The third session of the Semi-Annual Meeting was opened by President Jones on Saturday, June 24, at 9:30 a. m.

W. H. Driscoll reported on the conference held during the previous afternoon to discuss the elimination of the term square foot. He presented a resolution from the Committee on Nomenclature and moved its adoption as follows:

That this Society go on record as accepting 1000 Btu as the commercial unit of heat quantity and that it adopt the symbol *Mb* as the symbol of same and further that it adopt the symbol *Mbh* as an indication of the rate of heat flow per hour.

The motion was seconded and carried.

Ray C. Spitzley, Past President of the *Heating and Piping Contractors National Association* was introduced by President Jones and gave a few words of greeting from the Association.

Resolutions

The Resolutions Committee consisting of J. F. Hale, Chairman, H. L. Alt and P. M. O'Connell rendered its report, Mr. Alt making the presentation.

Inasmuch as one of the most successful meetings is now drawing toward a close,

Be It Hereby Resolved, that the Michigan Chapter should be highly commended for their courage in undertaking the responsibility for a meeting at this time and under the present conditions; and

Be It Further Resolved, that admiration be expressed for the efficient functioning of the various committees in providing such pleasing places of assembly, wonderful entertainments and the very efficient method of transportation, not forgetting the graceful services rendered to the ladies.

Be It Resolved, that it is the sense of this meeting that the hotel management should be commended for the excellent accommodations provided and the service they have rendered during the sojourn with them, also the local press for the recognition given to the convention.

Be It Resolved, that praise be expressed to the Program Committee for their excellent selection of papers presented at the technical sessions.

The resolutions were unanimously adopted.

W. R. Eichberg reported for the Nominating Committee, who announced the selections for Officers and Members of the Council who will serve in 1934.

The meeting adjourned at 11:45 a. m.

Conference on Nomenclature

At the Annual Meeting of the Society in Cincinnati, O., Pres. W. T. Jones appointed a Committee on Nomenclature to study the possibility of eliminating the term *square foot* and under the leadership of S. R. Lewis, Chicago, assisted by W. H. Carrier, P. D. Close, W. H. Driscoll, L. A. Harding, D. D. Kimball, J. R. McColl, C. L. Riley, W. A. Rowe, Perry West and A. C. Willard, several months were devoted to an exchange of views and the suggestion of terms that might be appropriate, practical, and acceptable.

In order to get the ideas of other organizations 18 allied groups were invited to participate in a Discussion of Nomenclature during the Semi-Annual Meeting of the Society at Hotel Statler, Detroit. The conference was held on Friday afternoon, June 23, and a most interesting discussion developed among the men in attendance.

At the suggestion of Dean F. Paul Anderson it was voted, "that the commercial unit of heat quantity shall be 1000 Btu." This, it was explained, is to supersede the term square foot and the steam and water units commercially used, 240 Btu and 150 Btu respectively.

During an extended and spirited discussion the following terms were considered:

Therm.—This is 1000 Btu in common use in America but it means 100,000 Btu in Great Britain. This was deemed a sufficient reason for rejecting the term therm.

Kb.—This was recommended by the Power Code Committee of the *American Society of Mechanical Engineers* but was not acceptable to many of the men in attendance at the meeting, since *K* in structural engineering is used to signify 1000 lb and there might be confusion.

M Btu—This was recommended by a number of the delegates as being self indicating as to its meaning.

Many other symbols were proposed and finally, on motion of W. H. Driscoll, by practically unanimous vote, it was decided that:

Mb shall be the symbol of 1000 Btu heat quantity.
Mbh shall be the symbol of 1000 Btu per hour.

The Chairman of the Society's Committee on Nomenclature, S. R. Lewis, expressed the opinion that repetition of these interesting and informal discussions on nomenclature might profitably be held at future annual and semi-annual meetings of the Society and the audience agreed with his proposal that such meetings be held.

Each of the cooperative organizations will be requested to accept and adopt the new unit and symbol as soon as expedient both in the daily practice of its members and in its literature.

The aim of future conferences will be to improve engineering nomenclature and through cooperative measures establish and develop uniform standards under the procedure of *A. S. A.*

PROGRAM SEMI-ANNUAL MEETING, 1933

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
HOTEL STATLER, DETROIT, MICH.
JUNE 22-24, 1933

(Eastern Standard Time)

Wednesday, June 21

6:30 P.M. Dinner and Council Meeting.

Thursday, June 22

8:30 A.M. Registration (Ballroom Floor—Hotel Statler)

9:30 A.M. Greeting by President Michigan Chapter and President Western Michigan Chapter

Response by Pres. W. T. Jones

Temperature Gradient Observations in a Large Heated Space by G. L. Larson, D. W. Nelson and O. C. Cromer

Indices of Air Change and Air Distribution by F. C. Houghten and J. L. Blackshaw

Testing and Rating of Air Cleaning Devices Used for General Ventilation Work by S. R. Lewis

Report of Committee on Atmospheric Dust and Air Cleaning Devices by H. C. Murphy, Chairman

11:45 A.M. Ladies Motor Trip and Luncheon—(Members who do not play Golf may accompany the ladies—tickets available at registration desk)—Bus leaves Bagley St. entrance Hotel Statler—Luncheon at Dearborn Inn followed by tour of Greenfield Village—Returning cars will leave at 5:00 P.M. for Meadowbrook Country Club

1:00 P.M. Meeting of Membership and Chapter Relations Committees, E. K. Campbell, Chairman
1:00 P.M. Golf Tournament—Meadowbrook Country Club
7:30 P.M. Buffet Supper, Music and Dancing—Meadowbrook Country Club

Friday, June 23

9:00 A.M. Meeting Nominating Committee
9:00 A.M. Meeting Committee on Ventilation Standards, W. H. Driscoll, Chairman
9:30 A.M. A Pipe Sizer for Determining the Sizes of Pipes and of Restricting Orifices in a Hot Water Heating System by L. A. Cherry
Application of the Eupatheoscope for Measuring the Performance of Direct Radiators and Convectors in Terms of Equivalent Temperature by A. C. Willard, A. P. Kratz and M. K. Fahnestock
Tests of Convectors in a Warm Wall Testing Booth (Part II) by A. P. Kratz, M. K. Fahnestock and E. L. Broderick
Report of Committee on Code for Testing and Rating Convection Heaters (Hot Water Section), R. N. Trane, Chairman
12:30 P.M. Ladies Luncheon (Hotel Statler)
1:00 P.M. Conference Committee on Nomenclature, S. R. Lewis, Chairman
1:00 P.M. Golf Tournament—(Research Cup)—Lochmoor Country Club Inspection of Industrial Plants (Register in Advance at Registration Desk)
2:00 P.M. Visit to Detroit Art Institute to view famous paintings and Rivera's Murals
5:30 P.M. Busses leave Hotel Statler for Grosse Pointe Yacht Club—touring Belle Isle enroute
7:45 P.M. Semi-Annual Banquet and Dance at Grosse Pointe Yacht Club (Informal)

Saturday, June 24

9:30 A.M. The Heat Conductivity of Wood at Climatic Temperature Differences by F. B. Rowley
Physiologic Changes during Exposure to Ionized Air by C. P. Yaglou, A. D. Brandt and L. C. Benjamin
Measurement of Air Flow through Grilles by L. E. Davies
Carbon Monoxide Distribution in Relation to the Heating and Ventilation of a One-Floor Garage by F. C. Houghten and Paul McDermott
1:00 P.M. Luncheon and inspection air conditioning exhibit—American Blower Corp. plant

COMMITTEE ON ARRANGEMENTS

L. L. McConachie, *General Chairman*

H. E. Paetz, *Vice-Chairman*

J. F. Johnston, Jr., *Representative of Western Michigan Chapter*

Finance Committee: E. H. Clark, *Chairman;* J. D. Cantwell and M. F. Mattingly.

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Registration Committee: E. E. Dubry, *Chairman;* E. H. Clark, S. S. Sanford, J. W. Snyder and G. H. Tuttle.

Transportation Committee: J. L. Fuller, *Chairman;* and H. F. Reid.

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No. 956

A. S. H. V. E. STANDARD CODE FOR TESTING AND RATING AIR CLEANING DEVICES USED IN GENERAL VENTILATION WORK

COMMITTEE

H. C. Murphy, *Chairman*,
J. J. Bloomfield,
Albert Buenger,
Philip Drinker,
Dr. E. V. Hill,
S. R. Lewis,
H. B. Meller,
F. B. Rowley,
Games Slayter,
Dr. S. W. Wynne

A. SCOPE

THIS Code is for the laboratory investigation and rating of devices used in general ventilation work for the sole purpose of removing solid impurities from the air. It is not intended for use in the rating of cyclones, dust separators or for devices used in the field of Industrial Hygiene. Ratings established under this Code are not to be confused with operating efficiencies in actual service—the dust concentration, the nature of the dust, the relative humidity and many other factors have a definite bearing upon the results obtained in actual service.

B. DEFINITIONS

1. **Standard Air** is air at 68 F, 29.92 in. of mercury barometric pressure and 50 per cent relative humidity. A variation of 10 F in temperature, 15 per cent in relative humidity and 1 in. barometric pressure shall be allowed but must be noted in the report of test.

2. **Air Capacity** (CFM) is the number of cubic feet of standard air per minute handled by the air cleaning device.

3. **Velocity** (FPM) is the air capacity per square foot of gross intake area of the air cleaning device, i. e., the linear feet per minute of the air passing through the filter proper.

4. **Limit Velocity** is the velocity of the air at which the accumulation of dust or liquid or parts of the filter medium in the air cleaning device com-

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Detroit, Mich., June, 1933, and adopted as amended.

mence to break away from it and reenter the air stream. It also may be the velocity of the air at which the resistance of the filter to air flow exceeds some stated value.

5. **Resistance** (R) is the resistance to air passage interposed by the filter itself and usually is expressed in inches of water. The following conversion table shows relative values of common methods of expressing air pressure:

Inches of Water	Ounces Avoirdupois per Sq In	Grams per Sq Cm
1	0.577	2.54
1.73	1	4.4
0.394	0.227	1

6. **Dust Concentration** is the amount by weight of dust contained in a specified volume of air, and usually is expressed in grams per 1,000 cu ft. The following conversion table shows relative values or common methods of expressing dust weight:

Grams	Pounds Avoirdupois	Ounces Avoirdupois	Grains
1	0.0022	0.0352	15.45
453.6	1	16	7000

7. **Dust Arrestance** (E) is the percentage relation which the dust concentration G_1 to leeward of the air cleaning device bears to the dust concentration G_0 at the same point in the same system when operated under the identical conditions but without the cleaning device. It is expressed by the formula $E = 1 - \frac{G_1}{G_0}$

The G_0 dust concentration shall always be determined *after* having made a test with the air cleaning device in place, so that the air handling capacity assuredly shall have been established.

8. **Reconditioning Power** is the energy required for operating the mechanism of an automatic air cleaning device and is not to be confused with the power required to overcome the resistance to air flow of the device.

9. **Standard Dust** shall be 50 per cent by weight of powdered lamp black, containing 97.5 per cent of free carbon minimum and having a bulking value of 3½ lb per cubic foot minimum. 50 per cent by weight of ashes from Pocahontas bituminous coal screened to pass 200 mesh.

10. **Dust Holding Capacity** is the amount by weight of standard dust which a non-automatic air cleaning device will retain before reconditioning is necessary.

C. PREPARATION OF THE AIR CLEANER

11. **The Device** to be tested shall be new and free from any foreign matter except the normal viscous coating or wetting, if any, with which it is designed to function.

For non-automatic viscous coated air cleaning devices the viscous coating shall have been allowed not less than 24 hours for draining after application.

For automatic viscous coated air cleaning devices the viscous coating shall be in normal condition of service.

D. CLASSIFICATION OF AIR CLEANING DEVICES

12. **Group A. Automatic Type:** In general all air cleaning devices which use power to automatically recondition the filter medium and maintain a non-varying resistance to air flow.

13. **Group B. Low Resistance Non-automatic Type:** Air cleaning devices for warm air furnaces, unit ventilating machines and similar apparatuses and installations in which a maximum of not more than 0.18 in. water gage is available to move air through the air cleaning device.

14. **Group C. Medium Resistance Non-automatic Type:** Air cleaning devices for systems in which a maximum of not more than 0.5 in. water gage is available to move air through the air cleaning device.

15. **Group D. High Resistance Non-automatic Type:** Air cleaning devices for the air intake of compressors, internal combustion engines and the like, where a pressure of 1.0 in. or more water gage is available to move air through the air cleaning device.

E. TEST CONDITIONS

16. **The Test Quantities** of air and of dust shall be as follows per square foot of air passing area exposed to windward for each type of air cleaning device.

a. Air: For devices of **Group** *A*—500 CFM. (10.5 grams dust per square foot per hour.)

For devices of **Group** *B*—250 CFM. (5.25 grams dust per square foot per hour.)

For devices of **Group** *C*—350 CFM. (7.35 grams dust per square foot per hour.)

For devices of **Group** *D*—350 CFM. (7.35 grams dust per square foot per hour.)

b. Dust: 0.35 grams (5.4 grains) of dust for each 1,000 cu ft of air passed through the device. The dust weights shown in brackets are on this basis.

The air and dust flow shall be under continuous observation and shall be maintained constant for each test period regardless of the tendency which may be manifest for the air volume to be curtailed due to increasing resistance caused by dust deposit.

17. **Time.** One hour shall be allotted to each observation of dust arrestance. Not more than six consecutive tests shall be made in any 24 hours.

18. **Basis of Rating for Non-automatic Air Cleaners.** A section having at least 2 sq ft face area of the air cleaning device shall be tested in the apparatus as set forth herein and standard air and dust in standard quantities

for each shall be employed. The dust holding capacity shall be determined by loading the air cleaning device with dust in hourly increments until such time as one or the other of the following conditions shall have been reached: (1) the arrestance shall have fallen below 85 per cent of maximum on performance curve or (2) the resistance shall have risen above:

0.18 in. for air cleaning devices of Group *B*
0.50 in. for air cleaning devices of Group *C*
1.00 in. for air cleaning devices of Group *D*

Having established the dust holding capacity, select the time during the test when there was deposited one-half of the volume of dust which had been deposited at the limit of dust holding capacity as defined previously.

At this selection point the air volume of course is known, and the resistance to air flow has been established. The average arrestance over the holding-capacity run, easily can be computed.

All non-automatic air cleaning devices rated under this Code shall disclose the following information:

———————— cubic feet per minute per square foot of face area.
———————— per cent average arrestance up to limit of holding capacity.
———————— inches of water resistance at 50 per cent holding capacity.
———————— grams holding capacity per square foot of face area.
———————— degrees Fahrenheit.
———————— per cent relative humidity.

Example 1

RATING TEST UNIT NO. 6-4 IN. THICK—GROUP C

Grams Dust per Sq Ft	Hour	Arrestance	Resistance Inches	CFM per Sq Ft
7.35	1	90.4	0.18	350
14.70	2	97.8	0.20	350
22.05	3	93.1	0.23	350
29.40	4	96.0	0.25	350
			*	
36.75	5	95.9	0.30	350
44.10	6	93.6	0.37	350
51.45	7	96.2	0.45	350
58.80	8	99.3	0.55	350
	Average	95.2		

* Intermediate rating point.

The average arrestance is determined from the arithmetical mean of the individual arrestances the dust-holding capacity as the total dust load multiplied by the average arrestance (95.2 per cent x 58.80 grams). The intermediate rating point is determined as the point at which one-half of the dust load has

been added: $\frac{58.80}{2} = 29.4$ grams. The average resistance is the resistance corresponding to this point.

This unit should therefore be rated at 350 CFM per square foot of face area with an average arrestance of 95.2 per cent at an average resistance of 0.25 in. with a dust holding capacity of 56.0 grams per square foot. (95.2 per cent of 58.80 grams.)

19. **Basis of Rating for Automatic Air Cleaning Devices.** Ratings shall be based on the manufacturer's smallest commercial unit having not more than 10 sq ft of face area, the longest dimension of which is not more than 5 ft.

The fully assembled device shall be tested in an apparatus as set forth herein and standard air and dust in standard quantities for each shall be employed. Dust shall be admitted in hourly increments and the re-conditioning mechanism shall be so adjusted that (1) the arrestance shall not fall below 85 per cent of its maximum and (2) the resistance shall not rise above 0.50 in W.G.

After the re-conditioning mechanism is adjusted the test shall be started and shall continue for not less than 48 operating hours for a continuously reconditioned device or two complete cycles for a periodically reconditioned device.

Record shall be made of the mean, constant rate of dust admission, the resistance to air flow and the arrestance of dust. The reconditioning power shall be measured and prorated to the air volume.

Rating tables for Automatic Air Cleaning Devices according to this Code shall contain the following information:

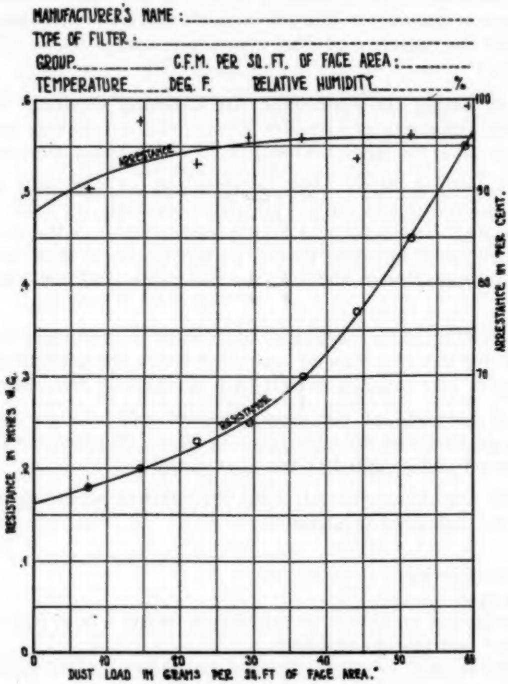
Method of Reconditioning————
Reconditioning cycle————
———— cubic feet per minute air capacity per square foot of face area.
———— per cent average arrestance.
———— inches W.G. average resistance (at 50 per cent of operating cycle if varying).
———— watts per 1000 cfm reconditioning power (whether this energy is used for moving the filter media or in pumping viscous coating or water or compressed air it should be reduced to watts).
———— degrees Fahrenheit temperature.
———— per cent relative humidity.

F. PERFORMANCE CURVES

20. Nothing in the foregoing is to be taken as interposing any objection to the furnishing by manufacturers of full performance curves covering operation with different kinds of dust, different concentrations, etc. They must be prepared so to do in order to demonstrate where required, the results obtainable in actual operation. If other than standard dust concentration is used, a note should be made as to its composition or rate of feeding on the performance chart.

The accompanying form of performance chart is approved—

PERFORMANCE CHART.



G. TEST EQUIPMENT

21. **Suitable Instruments** shall be provided for determining the condition of the air. The relative humidity is especially important as any increase in the moisture content of the air will cause a measurable increase in resistance to air flow through certain filter fabrics such as felt and the like. No reliable formulas are known for reduction of test data to standard conditions. For this reason all tests shall be made with standard air.

The air cleaning device shall be operated between two air chambers, a *mixing chamber,* which serves to mix the dust evenly with the air, and a *receiving chamber* of sufficient volume to distribute the air flow evenly over the device. The entrance or front end of the mixing chamber shall be equipped with an intake orifice serving to produce a highly turbulent flow in the front section of said chamber, whereby the mixing is effected. The rear end of the receiving chamber shall be connected to the suction side of a centrifugal blower by

means of a duct of suitable size. The air from the blower outlet shall be discharged outside the test room.

The *mixing chamber* and the *receiving chamber* shall be made of suitable material and be air tight. Both shall have equal cross section of dimensions at least equal to the dimensions of the intake or outlet areas of the air cleaning device to be tested. The *mixing chamber* shall be at least three times longer than its greatest cross section dimension. The intake orifice shall have an area not greater than one-quarter of the cross section area of the mixing chamber. The outlet of the mixing chamber shall have suitable means for attaching it to the intake of the air cleaning device.

The *receiving chamber* shall have a length not less than one and one-half times its greatest cross section dimension. The intake shall be provided with suitable means for attaching the air cleaning device. The outlet shall be

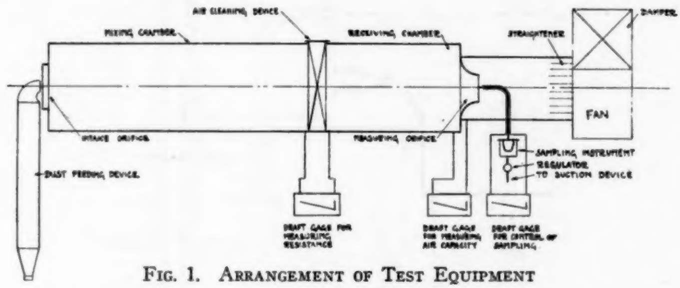


FIG. 1. ARRANGEMENT OF TEST EQUIPMENT

attached to the *duct*. The arrangement of the air chambers is illustrated in Fig. 1.

The *duct* shall be straight, of uniform area and circular section and of dimensions suited to the type of air measuring equipment used. In its blower end it shall be equipped with suitable straightening vanes or other arrangements for elimination of swirls from the blower.

22. **Air Flows** less than 2,000 cfm shall be measured by means of shaped orifices. Air flows greater than 2,000 cfm shall be measured either with such orifices or by the standard pitot tube method.

23. **Shaped Orifices** shall be of standard dimensions as shown in Fig. 2 and shall be positioned in the rear wall of the receiving chamber having their outlet openings inside the duct. The differential pressure shall be measured by means of one or more static tubes on each side of the orifice. The air passing through the orifice shall be computed from:

$$\text{CFM} = 4000\,A\sqrt{VP}$$

A being the total outlet area of the orifice or orifices and *VP* the differential pressure. The area *A* shall be of such size that the velocity through it shall be at least 3000 linear feet per minute when the Air Cleaning Device operates at Rated Air Capacity.

24. **If a Pitot Tube** is used it shall be of standard construction, as shown in Fig. 5. The duct shall be at least 10 diameters in length and of such cross

section area that the mean air velocity in it is at least 3000 fpm. The readings shall be taken in a plane normal to the axis of the duct and at a distance from the receiving chamber of three-quarters of the length of the duct. (Fig. 5.)

Not less than 20 readings shall be taken (not less than two traverses of ten readings each along diameters at right angles to each other) at the centers of

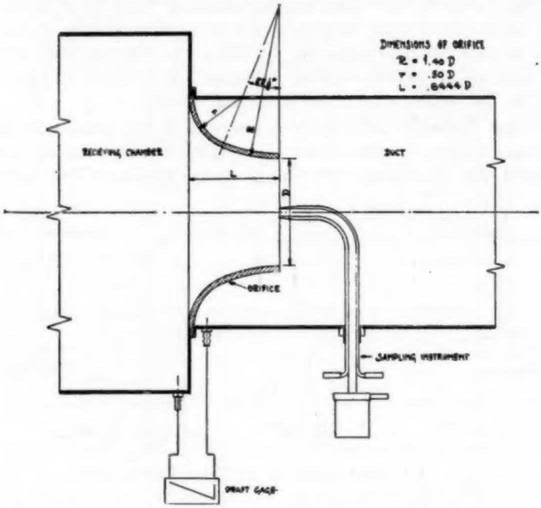


FIG. 2. STANDARD SIZE AND POSITION OF ORIFICES

five equal annular areas. (Fig. 5.) The velocity V shall be computed from the Velocity Pressure reading VP by means of the formula.

$$V = 4000\sqrt{VP}$$

The air volume flowing through the duct shall be computed from:

$$\text{CFM} = AV_m$$

A being the cross section area of the duct and V_m being the arithmetical mean value of all the velocities measured.

25. **The Pressure Measuring Instruments** may be of any suitable type and shall be calibrated before each test by means of a vertical U-tube containing pure water.

26. **The Resistance** shall be measured by means of one or more total pressure tubes on each side of the air cleaning device. These tubes shall be positioned as close to the air cleaning device as possible. If several are used on each side they shall be placed in one plane normal to the air flow.

In cases where the dimensions of the air chambers are exactly equal to the dimensions of the inlet and outlet openings of the air cleaning device static tubes may be employed instead of the total pressure tubes.

27. **Dust Admission** shall be accomplished by means of an automatic dust

feeding device capable of feeding dust at an approximately uniform rate. It shall be adjusted to liberate approximately 0.35 grams (5.4 grains) of dust for each 1,000 cu ft of air passed through the device.

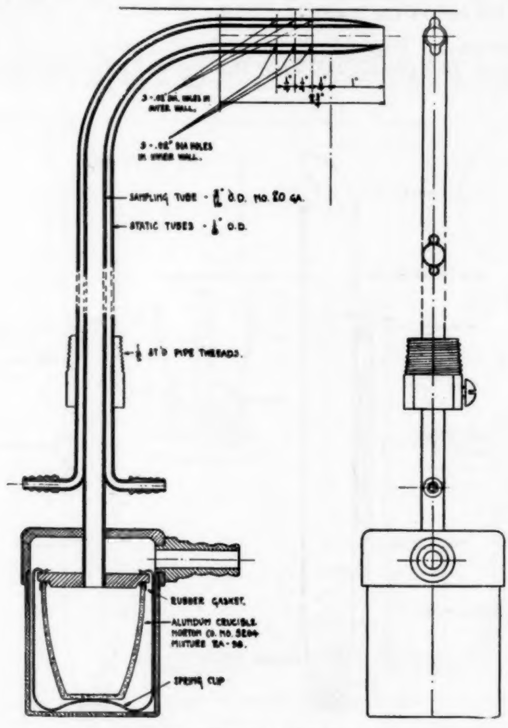


FIG. 3. CONSTRUCTION OF SAMPLING INSTRUMENT

The dust shall be dried for at least one hour at 220 F. After drying it shall be divided into hourly charges computed as follows:

air per minute × 0.021 = grams per hour

Not less than two preliminary hourly test runs shall be made to establish the air volume and the dust feeding adjustment. A balance capable of weighing to 0.0001 gram shall be provided for weighing the dust charge.

Example 3

If it is proposed to clean 220 cfm per square foot of face area the dust should be divided into uniform charges each capable of supplying this volume of air for an hour with its quota of dust.

Thus: *per square feet* 220 × 0.021 = 4.62 grams.

If the dust charge should have been released prior to the expiration of one hour, the testing device shall continue to be operated without further admission of dust until the end of the hour.

Dust admission per hour for 20 in. x 20 in. testing unit, at 400 cfm, 8.4 grams; 600 cfm, 12.6 grams; 800 cfm, 16.8 grams; 1000 cfm, 21.0 grams.

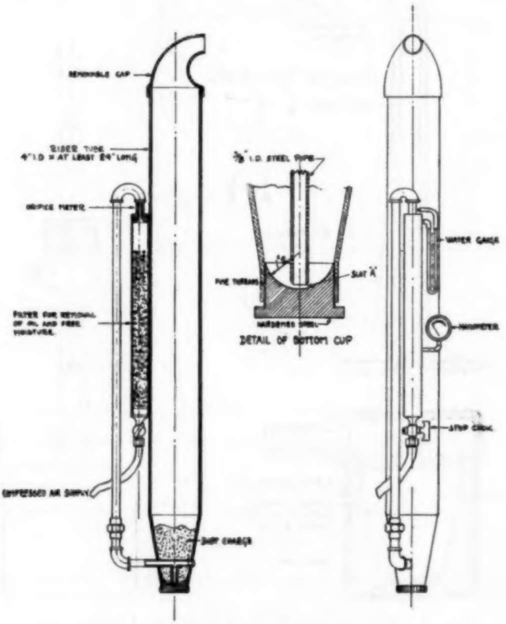


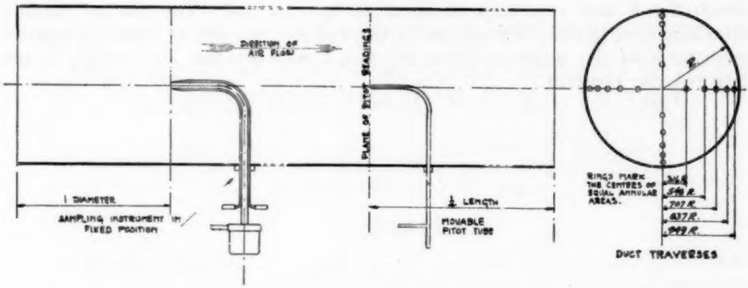
FIG. 4. APPROVED DUST FEEDER

28. **The Dust Concentration** shall be measured by means of a clay crucible which shall be operated in a standard sampling instrument, and weighed accurately before and after each test. Immediately preceding each weighing the crucible shall be dried in a desiccator for at least one hour. An analytical balance capable of weighing to 0.0001 gram shall be provided for this purpose.

The sampling instrument shall be constructed in accordance with Fig. 3, and shall be operated in a vertical position with the crucible facing upwards. The suction pipe shall be connected to a suitable suction device of ample capacity to overcome the resistance of the tube and the crucible. The connecting line shall be provided with a suitable valve or damper for regulation of the *sample volume* of air.

For proper operation of the sampling instrument the sample must be taken in a moving stream of at least 3000 fpm velocity. The air velocity within the sampling tube must be equal to the velocity of the air stream. Such will be the case when the differential pressure across the static tubes of the sampling instrument is zero.

An inclined differential draft gage capable of reading to 0.01 in W.G. shall be employed across these static tubes. This gage need not be calibrated.



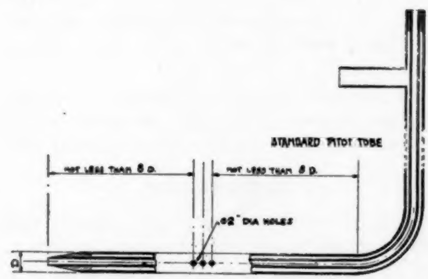


FIG. 5. PITOT TUBE STANDARD SIZE

29. **The Power Consumption** shall be determined by means of a calibrated electric motor and suitable electrical instruments, preferably including a watt-hour meter.

30. **Dust Feeder:** An approved dust feeder is shown in Fig. 4. It consists of a vertical tube having a nozzle on its top and a dust chamber on its bottom. Compressed air enters through a packed filter and through the slot *A* forces the dust upward through the tube and then through the cap orifice into the mixing chamber of the testing apparatus.

A pressure gage and an orifice meter with a water gage shall be provided for measuring the pressure and volume of the compressed air supply. If the compressed air source is at high pressure an automatic pressure regulating valve in the compressed air supply line should be supplied. This device in service should be operated so as to provide a reasonably uniform rate of dust injection into the air stream.

NOTE: Dust having been introduced in the device by removing the cap forms a bed in the lower part of the device. The compressed air entering through the slot *A* produces an agitation in this bed of dust and the violence of the agitation depends on the air velocity in the slot. In order to obtain a uniform discharge of all particle sizes contained in the dust the air volume should be so adjusted that the air velocity in the tube will be greater than the velocity of fall of the heaviest particles. For such purpose the water gage of the orifice meter should be graduated to read this velocity directly. With standard test dust a velocity of about 20 fpm is used. The rate of feeding should be controlled by adjustment of the width of the slot *A* and simultaneous adjustment of the supply pressure in such a way that the air velocity in the tube remains constant.

No. 957

A. S. H. V. E. STANDARD CODE FOR TESTING AND RATING CONCEALED GRAVITY TYPE RADIATION (HOT WATER CODE)

COMMITTEE: R. N. Trane, *Chairman;* E. H. Beling, H. F. Hutzel, A. P. Kratz, O. G. Wendel, J. H. Holton, M. G. Steele, R. F. Connell

A. PURPOSE

1. Object of Code. The object of this code is to provide a practical method for testing and rating convectors which may be readily used by manufacturers, and which will provide the necessary information for the heating engineer in the design of the heating plant. This code describes a boiler load method of rating.

2. Definition. The term convector as used in this code shall apply to any type of radiator installed within a wall, casing, or enclosure, having an inlet and outlet for the circulation by gravity of the air in the room to be heated.

B. APPARATUS

3. Purpose of Test Booth. The purpose of the test booth shall be solely to maintain constant and uniform conditions about the convector.

4. Description of Test Booth. The test booth shall be 12 ft x 15 ft in floor area, and shall have a ceiling height of 9 ft. A tolerance of $\pm$ 10 per cent in any dimension is permissible. The floor of the booth shall be set up in a larger room (See par. 5) at least 1 ft above the room floor. One side of the booth (either a 12-ft or a 15-ft side) shall be open, with a shield projecting down from the top at least 1 ft. The walls and ceiling shall have a smooth, close surface, and shall be painted with oil paint. The air in this test booth shall be free from draft.

5. Room Enclosing Test Booth. The test booth shall be set up in a larger room and in such a position as not to be exposed to the direct rays of the sun. The distance between any test booth wall and the wall of the surrounding room shall not be less than 2 ft and the ceiling of the test booth shall not be less than 1 ft from the ceiling of the larger room. The air temperature in this larger room shall be taken at three sides of the test booth at a level of 5 ft from the floor of the larger booth at a distance of 12 in. from the test booth walls and shall not show a variation to exceed $\pm$3 deg (F) during the course of a test.

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Detroit, Mich., June, 1933, and adopted as amended.

6. Location of Convector in Booth. The convector shall be set up in accordance with the manufacturer's specifications, adjacent to the wall opposite the open side of the test booth.

7. Convector Casing. The convector shall be placed either in a casing with grilles as furnished by the manufacturer, or if a casing is not furnished, it shall be placed in a box made of material simulating the catalog specifications.

8. Apparatus. The apparatus shown in Fig. 1 for Steam Method and Fig. 2 for Electrical Method of testing shall consist of the following:

Steam Method

A. Constant Head Tank
C. Water Control Needle Valve
E. Water Manometer
F. Convector Connection
G. Convector
H. Steam Water Heater
J. Scales
K_1 and K_2 Location of Thermometers or Thermocouples
L. Water Weighing Can
M. Sump Tank
O. Constant Head Tank Vent Valve
P. By-pass Valve
R. Steam Separator
S. Mercury Manometer
T. Steam Reducing Valve
U. Centrifugal Water Pump
V. Water Control Valve
W. City Water Valve
Y. Condensate Level Valve

Electrical Method

A. Constant Head Tank
B. Thermostat
C. Water Control Needle Valve
D. Contactor
E. Water Manometer
F. Convector Connection
G. Convector
H. Electric Water Heater
J. Scales
K_1 and K_2. Location of Thermometers or Thermocouples
L. Water Weighing Can
M. Sump Tank
O. Constant Head Tank Vent Valve
P. By-pass Valve
U. Centrifugal Water Pump
V. Water Control Valve
W. City Water Valve

9. Constant Head Tank. A tank *A* vented to the atmosphere shall be used having a minimum capacity of 7 gal to the water line established.

10. Water Heater. An insulated water heater *H* located outside of the test booth, either steam type or electric type in accordance with the method of testing chosen, shall be used having sufficient heating capacity to heat the necessary amount of water to the proper temperature for the test undertaken. In the case of the electric heater, the electric input and the outlet water temperature (See Fig. 2) shall be controlled by means of a thermostat *B* and a contactor *D*.

When steam heater is used, it shall be connected as shown in Fig. 1, the water to be heated shall pass through the coil and steam shall occupy surrounding space.

Steam shall be supplied at a constant pressure, through a reducing valve *T* (Fig. 1).

11. Sump Tank. A sump tank *M* of minimum capacity of 80 gal shall be conveniently located as near the scale *J* and weighing tank *L* as possible.

12. Centrifugal Pump. A centrifugal pump *U* shall be provided, having sufficient capacity to at all times pump more water to the constant head tank *A* than is delivered to the convector. It shall be permissible to supply the constant head tank *A* with water delivered directly from the water main where the heater *H* has sufficient heating capacity to heat the water to the required temperature. In this case the sump tank *M*, centrifugal pump *U*, and valve *V*

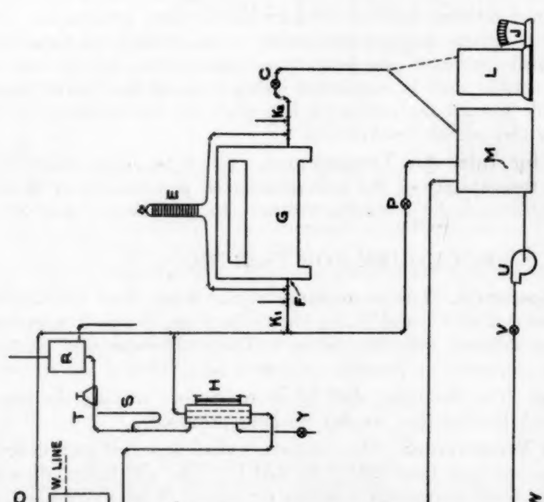


FIG. 1. STEAM METHOD. DIAGRAM SHOWING CONNECTIONS AND EQUIPMENT FOR SUPPLYING HOT WATER TO THE CONVECTOR AND MEASURING SAME

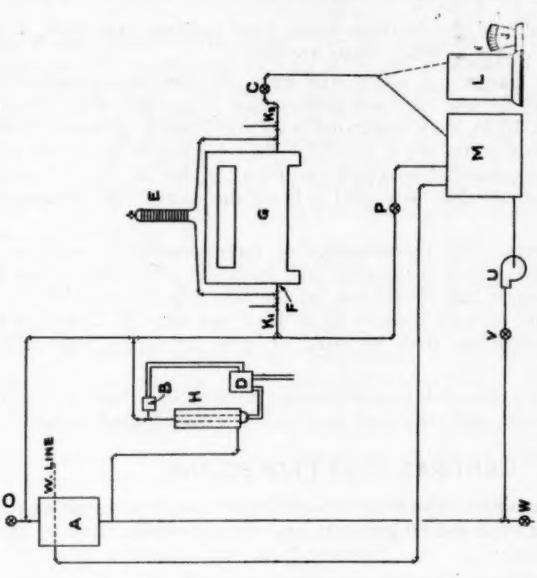


FIG. 2 ELECTRICAL METHOD. DIAGRAM SHOWING CONNECTIONS AND EQUIPMENT FOR SUPPLYING HOT WATER TO THE CONVECTOR AND MEASURING SAME

TO SIMPLIFY ILLUSTRATIONS THESE DIAGRAMS SHOW ALL APPARATUS IN A SINGLE PLANE. IT IS ESSENTIAL THAT ALL NECESSARY EQUIPMENT BE LOCATED AS NEAR TEST BOOTH AS POSSIBLE

will not be required, and the overflow water from constant head tank *A* shall be delivered to the sewer or other waste receiver.

13. Differential Gauge. A differential gauge *E* shall be an inclined and inverted U-tube with parallel tubes not greater than ⅛ in. bore, with inclination of at least 1 in. in 10 in. with calibrated scale for reading pressure drops to 0.01 in. The pet cock on the top is to release the collection of air in the gauge. This gauge shall be connected to supply and return piping as close to test unit as possible. Enough air shall be vented to bring the water level in gauge to a convenient height.

14. Thermometers. The thermometers or thermocouples (hereinafter designated *thermometers*) used in the tests shall be calibrated by comparison with a standard thermometer and in the case of the thermometers used at K_1 and K_2 shall be suitable to read directly to at least one-fifth of 1 deg (F) or equivalent. Thermocouples shall be made of wire not larger than 0.02 in. diameter.

15. Scales. The scales shall be a beam type, reading to at least 1 oz. They shall be checked before and after each test with a standard check weight.

GENERAL TEST PROCEDURE

16. Relative Humidity. The relative humidity at time of test shall be within the limits of 25 per cent and 60 per cent, and the air motion shall be limited to natural circulation.

17. Air Temperature. The convector shall be tested with an inlet air temperature of not less than 60 F or more than 75 F, this temperature to be measured with thermometers placed 12 in. apart (minimum of three) throughout the length of the inlet opening midway between the top and bottom, and 3 in. in front of the inlet. These thermometers shall be shielded against radiation by semicircular shields made of pieces of thin bright metal, 3½ in. high by 3 in. long. These shields shall be suspended independent of the thermometers, with the convex side toward the convector and with the thermometers at the center of the semicircles on the concave side.

18. Correction for Inlet Air Temperature. When the inlet temperature differs from 65 F, the capacity of the convector under test conditions shall be converted to the capacity under desired conditions, as explained in par. 27.

PROCEDURE FOR TESTING

19. Water Temperature. The temperature of the water shall be measured by thermometers located at K_1 and K_2 as shown in Figs. 1 and 2 with bulbs immersed in direct contact with the water. These thermometers shall be located as near the convector as possible (within 6 in.).

20. Air Venting. The convector shall be vented before starting the test by means of an air cock through the regular opening provided.

21. Inlet Water Temperature. The convectors shall be tested with different water temperatures, ranging from 100 F to 220 F. The maximum allowable variation in the inlet water temperature during the test shall not exceed ± 1 deg (F) above or below the test average. Variation in the water rate to be not more than 2 per cent.

22. Water Temperature Drop. The convector shall be tested with a water temperature drop of 20 deg, with an allowable variation of ± 1 deg (F).

23. Start of Test. The temperature of the water in sump tank *M* shall be brought to a suitable temperature so that heater *H* can conveniently furnish water of the proper test temperature (par. 21) to the convector. Needle valve *C* shall be adjusted to pass the required quantity of water to give the desired water temperature drop as indicated by the thermometers located at K_1 and K_2 (par. 22). The temperature of the inlet air to the convector shall be as specified in par. 17.

The test shall start when all conditions of water temperature and inlet air temperature are reasonably constant as heretofore specified and after such conditions have existed for at least one hour. In the case of the Steam Method, the test shall start with the water in the gauge glass of steam heater *H* at the level required to give the required test temperature at K_1 and this level shall be marked by a string on the gauge glass, and maintained at this level throughout the test. In the case of the Electrical Method, the test shall start with the heater circuit adjusted to maintain correct water temperature.

The by-pass valve should be adjusted in conjunction with the water heater to obtain constant conditions prior to test.

24. Readings. The readings of water temperatures, inlet air temperatures, friction pressure drop and water weight, shall be taken simultaneously (within the maximum allowance time of 30 seconds).

25. Duration. The test shall continue until ten sets of readings within prescribed limits as indicated in par. 21, 22, and 24 have been obtained.

COMPUTATION PROCEDURE

26. Output Under Test Conditions. *Steam Method—Electrical Method.* The hourly output of the convector under test conditions shall be determined by the following formulae and procedures:

$$H = W \times (\theta_1 - \theta_2) \times \frac{3{,}600}{t}$$

Where

$H =$ Convector output—Btu per hour
$W =$ Scale weight of water handled for duration of test—pounds
$\theta_1 =$ Average inlet water temperature at K_1 in degrees Fahrenheit
$\theta_2 =$ Average outlet water temperature at K_2 in degrees Fahrenheit
$t =$ Duration of test—seconds

27. Correction Factor. The correction factor *C* to convert hourly test output to output under desired conditions of standard air temperature of 65 F and standard average water temperatures of 170, 190, 210, and 230, corresponding to the required inlet temperatures of 160, 180, 200, and 220, shall be determined by use of the following formula:

$$C = \left[\frac{\theta_s - 65}{\frac{\theta_1 + \theta_2}{2} - \theta_3}\right]^{1.3}$$

$\theta_3 =$ Average test inlet air temperature.
$\theta_s =$ Standard average water temperature (170, 190, 210 or 230).

28. Output Under Standard Conditions. The output under desired conditions shall be in Btu per hour, determined by use of the following formula:

$$H_s = CH$$

Tables should show inlet water temperatures of 160, 180, 200, and 220 F with 20 deg drop for each width of heater for various lengths and heights, with resulting Btu output and pressure drops, as suggested in sample table shown in Fig. 3. Ratings for larger sizes must not be made by extrapolating from results on smaller sizes. However, ratings for intermediate sizes may

TYPICAL CAPACITY TABLE

WIDTH = 5″

The ratings given below are in Btu per hour based on following condition:

Air inlet temperature = 65 F
Water inlet temperature = 180 F
Water outlet temperature = 160 F

Stack Height Inches		Length of Heating Element in Inches				
		20	30	40	50	60
20	Btu	1960	2940	3820	4800	5780
	Friction	10.0	22.0	35.0	58.0	75.0
30						
40						
50						

Btu is Btu per hour (H).

Friction pressure drop through the heater in mil-inches of water.

The rate of water flow in pounds per hour for any radiator is approximately equal to $\frac{H}{20}$ or $\frac{H}{\text{Temperature Drop}}$.

FIG. 3. TYPICAL CAPACITY TABLE

be obtained by interpolation from the results on adjacent sizes. A sufficient number of sizes should be tested to definitely determine the curve expressing the relation between capacity and size. Friction pressure drop shall be in thousandths of inches (mil-inches).

29. Description of Grilles. The manufacturer shall state the free air area, type and location of grilles upon which the ratings are based. He shall also state the distance from the bottom of the convector to the top of the outlet grille which shall be designated as the stack height.

30. Sources of Error in Convector Testing. The major sources of error, in addition to accidental mistakes, are as follows:

1. Incomplete venting of air from convector.
2. Excessive air currents within the test booth.
3. Wet and/or insufficient insulation.
4. Calibration of thermometers, scales, thermocouples.
5. Starting test before equilibrium is obtained in system.

No. 958

TEMPERATURE GRADIENT OBSERVATIONS IN A LARGE HEATED SPACE

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Madison, Wis.

This paper is the result of research, sponsored by the American Society of Heating and Ventilating Engineers and conducted by the Mechanical Engineering Department of the University of Wisconsin

RESEARCH data concerning stratification of air in buildings are lacking and, when confronted with an unusual type of building, the designer often finds himself literally at sea in selecting inside temperatures upon which to base heat loss calculations.

According to the The A. S. H. V. E. Guide, it is the common practice of engineers to allow 2 per cent per foot of height above the breathing line in establishing the probable air temperatures at any given level for a direct radiation system, and not less than 1 per cent per foot above the breathing line in arriving at the probable air temperatures at any given level when a combined heating and ventilating system is used.

While values thus obtained are sufficiently accurate and undoubtedly safe in most cases, this method of calculation leads to erroneous and even absurd results when applied to large open rooms with unusually high roofs or ceilings. It was for the purpose of obtaining data on this subject that this research was undertaken.

The present study of temperature gradients has been carried on in the University of Wisconsin Field House, a building which is used primarily for basketball games and for indoor track. Fig. 1 shows the exterior of the building. The interior dimensions of the main part are as follows: width 193 ft, length 227 ft, height 97 ft. In addition to the main space, there is an additional space formed under the bleachers of the adjacent football stadium. The ⅛ mile running track goes through this space and the fan rooms, locker and shower rooms are also located here. The total internal volume of the building is about 3,500,000 cu ft.

The plans for the building provide for an ultimate seating capacity for

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basketball purposes of about 11,500. This capacity is to be secured by the use of two balconies accommodating 8,000 and wooden bleachers seating 3,500. The upper balcony has not yet been built. With temporary bleachers built up from the playing floor, the largest capacity secured up to the present is about 6,000. Fig. 2 shows a plan and elevation of the building. The lower balcony extends from the 13 ft elevation to the 25 ft elevation. The second balcony will occupy an elevation from 29 ft to 50 ft. Fig. 3 shows the general relation of the balconies and the playing floor. The support members for the upper balcony show in this view. The playing floor is large enough for two practice courts cross-wise of the building. For competitive games, the court length-wise of the building is used as shown in the photograph. Bleachers are erected

FIG. 1. VIEW OF FIELD HOUSE AT THE UNIVERSITY OF WISCONSIN

on all sides of the playing floor for extra seating capacity. The floor is made up of bolted sections which are dismantled and stored at the end of the basketball season. The support is on wood posts which rest on concrete blocks a few inches under the surface of the dirt floor. The dirt floor is used for spring football and baseball practice as well as for indoor track meets.

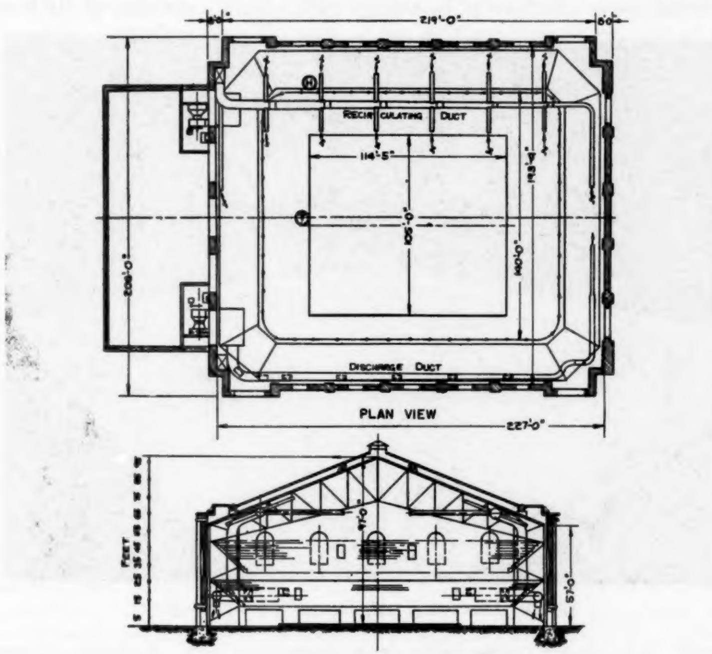
The exterior walls are of Madison sandstone backed up with poured concrete. The building is well lighted by large windows. The roof construction is 2 in. plank, 1 in. of cork and tile. Double-glazed sky-lights are located in the center of the roof as shown in Fig. 1.

DESCRIPTION OF HEATING AND VENTILATING SYSTEM

During games, it is desired to keep the playing floor relatively cool. The spectators require a somewhat higher temperature. Since they are located in balconies at a higher elevation, the natural tendency for heated air to rise, aids this desired differential heating. However, the spread of the balconies over an elevation from 13 ft to 50 ft makes the problem of securing uniform tempera-

ture for all spectators serious. During periods of inoccupancy, it is desired to keep temperatures at all elevations low for the sake of economy.

In order to keep roof temperatures within reasonable limits and to fulfill the desired conditions at the balconies and playing floor, a fan system of heating was decided upon. Since the building is used only a few hours at a time and the air volume per person is large, ventilation at all times of occupancy is not required. At times the moisture content may be high due to evaporation



SECTION THROUGH CENTER

FIG. 2. PLAN AND SECTION OF THE FIELD HOUSE SHOWING THE RECIRCULATING AND DISCHARGE DUCTS

from the dirt floor when sprinkled, in addition to that given off by the occupants. This makes it desirable to have available an outside air supply. Further, in mild weather, the heat given off by the occupants may need to be dissipated by supplying outside air.

The heating and ventilating system is in two duplicate parts and is shown in Fig. 2 in plan and elevation. The two fans have each a measured capacity of 55,000 cfm. The air supply may be taken from outside entirely or the system may recirculate a certain proportion or all of the air. The resistance of the outside air supply system is equal to the resistance of the recirculating duct system. The recirculating inlets are located near the roof. For each fan, there

are six 30 in. inlets located 58 ft from the floor, six 24 in. inlets at an elevation of 73 ft and three 30 in. inlets at each end of the building at an elevation of 66 ft.

Three tempering coils each consisting of 1,050 lineal feet of ⅝ in. finned copper tubes two rows deep are located side by side in each fan chamber. The air discharge is from a trunk line located under the lower balcony. There are twelve 29 in. by 56½ in. outlets directed downward in each of the two discharge ducts. Each outlet is equipped with a heater consisting of 212 lineal

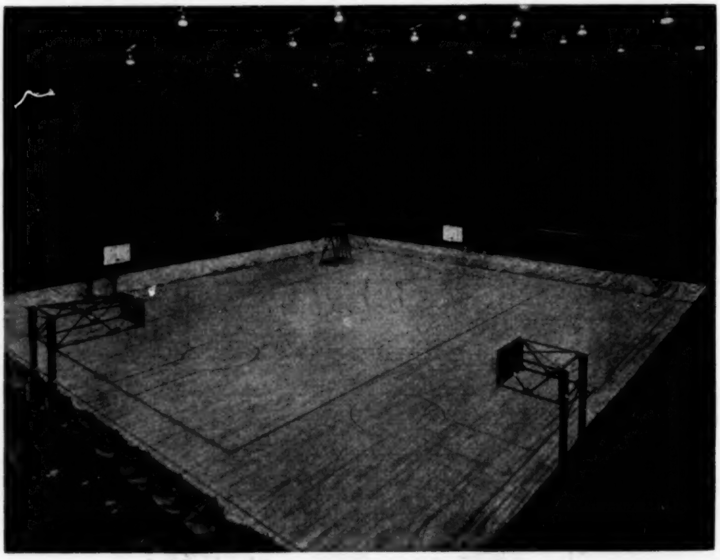


FIG. 3. INTERIOR VIEW, SHOWING PLAYING FLOOR

feet of ⅝ in. finned copper tubes placed three rows deep. Each one of these outlet heaters is under control of a thermostat placed on the structural columns above the lower balcony at an elevation of about 25 ft. These thermostats are set to maintain the temperature desired in the balconies. The action of the system when recirculating is to draw the warm air from the upper levels near the roof, heat it to the desired temperature at the various outlets and allow it to rise up through the open spaces of the balcony.

When arranged to take an outside supply, the air is heated to about 55 F in the tempering coils and is then heated to the desired balcony temperature at each individual outlet. Twenty 36 in. diameter roof ventilators are provided in four rows, each row having an independent manual damper control.

No direct radiation is used except in the shower, toilet and locker rooms which total 1,072 sq ft. The supply and return mains of the vacuum steam system are not covered and constitute direct radiating surface. The area of steam supply mains and branches is 1,524 sq ft. The lights over the playing

floor also have considerable capacity as heating elements. Over the playing floor, there are thirty 1,500 watt lights and twelve 1,000 watt lights totalling 57,000 watts. These are at an elevation of 58.5 ft to 73 ft.

Nature of Observations

Observations under various conditions as to occupancy, weather and operation of system have been made during the three seasons the building has been in use. In order to obtain the temperatures from floor to roof, ten resistance thermometers were hung from the peak of the roof structure at one end of the playing floor. The first thermometer was placed at the 5 ft level and the last one at the 95 ft level with 10 ft spacing between them. Readings were taken by means of a Wheatstone bridge arrangement through a 10-point switch. The thermometers were checked against mercury thermometers by immersion in a water bath. Approximately 1,000 ft of duplex lighting cord was used to make the circuits. The location of the thermometer cable is shown on the plan view marker as *T* on Fig. 2. Relative humidities were determined directly by a recording instrument. The wet and dry bulbs for this instrument were located at the point marked *H* on Fig. 2 on a beam directly below the lower part of the balcony in line with the resistance thermometer cable. Checking of temperatures showed that the values here generally corresponded closely to the temperatures obtained at the same elevation of 15 ft in the center of the building.

The steam used was measured by a recording steam flow meter placed in the main supplying the building. The amount recorded included a certain amount of pipe line loss and also the steam used to heat the shower water.

Observations have been taken during all of the basketball games of the past three years. In addition, readings have been taken during times of non-occupancy such as on zero days to observe gradients during still air conditions. The results of these various observations are shown in Figs. 4 to 9. The conditions of operation were established by the janitor, who decided when to operate the fans recirculating or with outside air and whether some or all of the roof vents should be open. In general during the first two seasons a temperature of 55 F was desired at the playing floor. During the past season, this desired temperature has been increased to 60 F or slightly above. The balcony thermostats have been set for 65 F during all three heating seasons.

Methods of Operation of System

There are several methods of operation of the system and each has been used during some of the games. Fig. 4 shows results of operation in mild weather and with a large attendance without the fans running, that is, with gravity operation. Fig. 5 presents the observations using recirculation in fairly cool weather. Fig. 6 shows results with the use of outside air and partial recirculation during a game with a small attendance in near zero weather. During this game, all outside air was used for a period. Fig. 7 shows similar operation in mild weather and with a larger attendance. During all of these games, the roof vents were open.

In addition to the results of observations during occupancy shown in Figs. 4 to 7, results obtained during cold weather and no occupancy are shown in Figs. 8 and 9. The main purpose of these latter figures is to show the

temperature gradient under severe heating conditions and with no disturbance of stratification tendencies. In Fig. 8 the influence of the starting of the

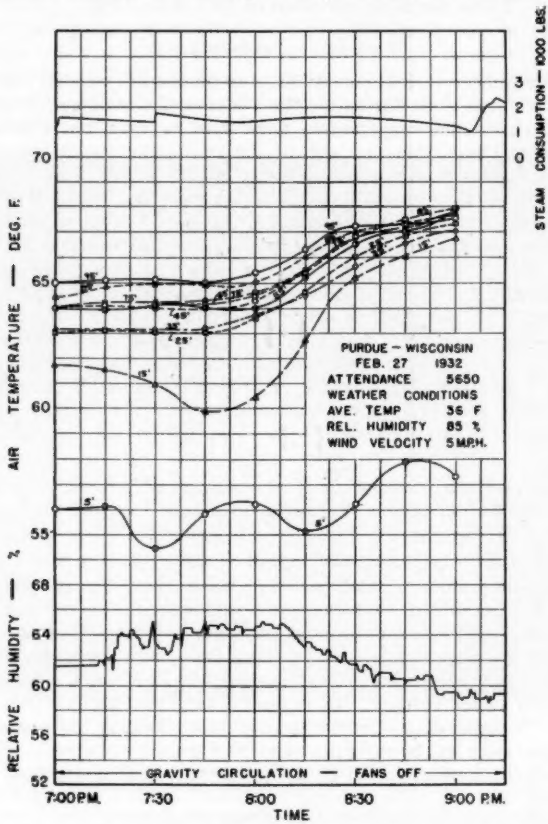


FIG. 4. RECORD OF TEMPERATURES, HUMIDITIES, AND STEAM CONSUMPTION WITH GRAVITY CIRCULATION—FANS NOT RUNNING

fan on the still air gradient is studied. In Fig. 9 the gradients observed on three below-zero days are shown.

GRAVITY CIRCULATION

During gravity operation of the system, the circulation through the ducts is in the opposite direction to that obtained with fan recirculation. The rate of air circulation depends upon the extent of the heating of the air as it travels upward through the ducts. The air enters at the discharge outlets below the

lower balcony and is further heated by the outlet heaters if the balcony thermostats are calling for heat. Since the capacity of the system is very much limited under gravity circulation, under all but mild weather conditions steam

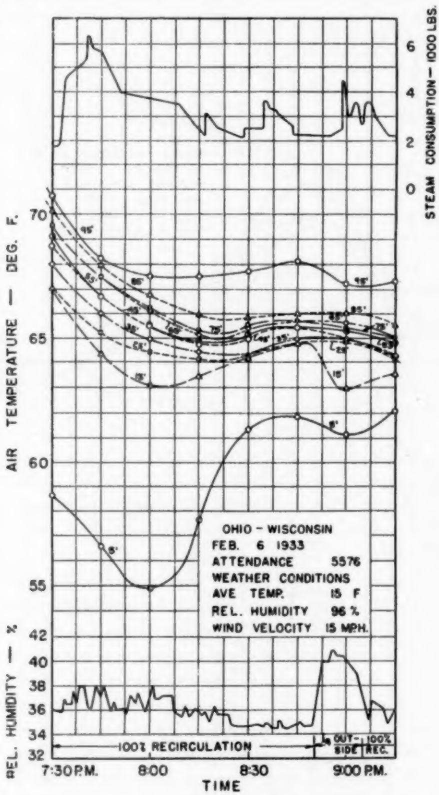


Fig. 5. Record of Temperatures, Humidities, and Steam Consumption Using Totally-Recirculated Air

would be present in these outlet coils. In mild weather only one of the three tempering coils is kept on. The air, then, on its upward travel receives but little heat in the fan rooms since its flow is divided between the area consisting of the by-pass and the two tempering coils not supplied with steam, and the area of the one tempering coil which is supplied with steam.

Fig. 4 shows the results secured with such circulation when the outside temperature was 36 F and when a capacity crowd of 5,650 was present. These conditions would seem to indicate that outside air might well be introduced by fan operation for cooling. The heating capacity of the system standing idle

was sufficient to bring the average temperature of the balcony elevations to about 63 F as shown at 7:00 p. m. The air immediately above the playing floor was about 56 F in temperature which was satisfactory. The rise in

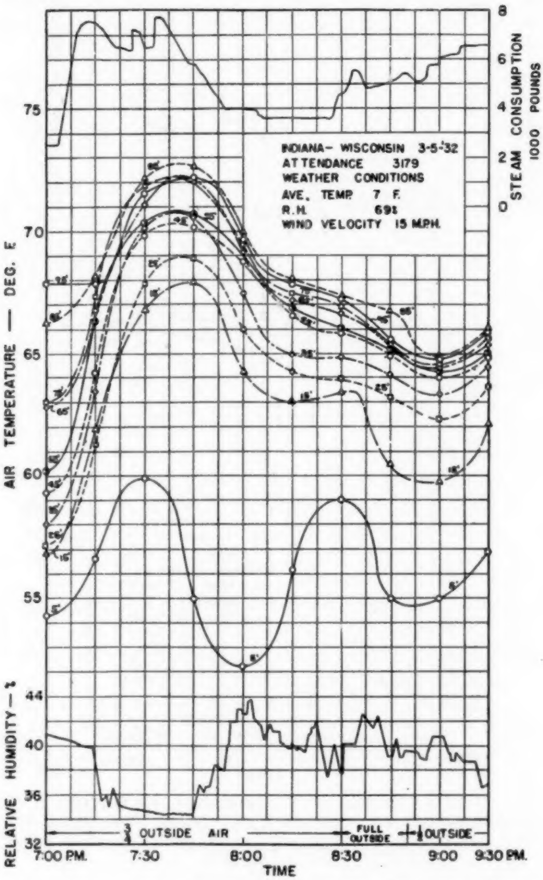


FIG. 6. RECORD OF TEMPERATURES, HUMIDITIES, AND STEAM CONSUMPTION USING OUTSIDE AIR IN COLD WEATHER

playing floor temperatures from 7:30 to 8:00 p. m. indicated that outside air was needed for cooling. Some windows were opened at about 8:00 p. m. which helped to keep the rise at the playing floor to a total of 3 F. The rise at the 15 ft level was 7 F. The rise at the balcony levels which are from 13 ft to 50 ft was about 4.0 F. The variation in temperature from highest to lowest point of the balconies was 3.8 F at 8:00 p. m. and decreased to 1.0 F at

9:00 p. m. This small difference of 1.0 F appears to be due to the uniform emission of low temperature heat by the large audience. Since the balcony thermostats are set at 65 F, the outlet heaters probably shut off sometime

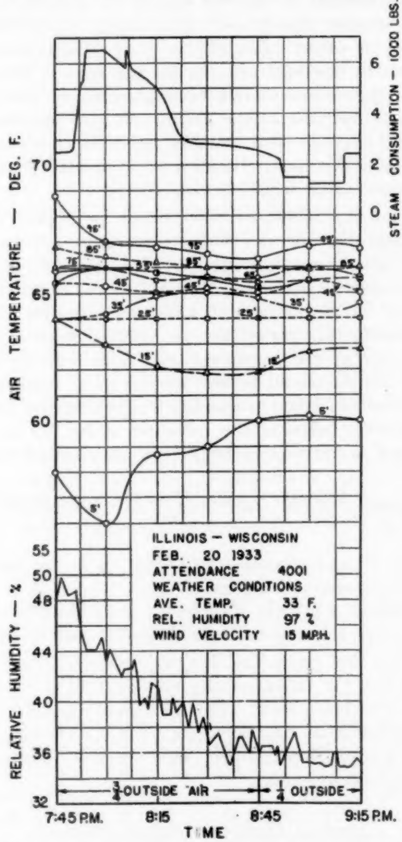


FIG. 7. RECORD OF TEMPERATURES, HUMIDITIES, AND STEAM CONSUMPTION USING OUTSIDE AIR IN MILD WEATHER

after 8:00 p. m. The highest temperatures are not at the roof but about mid-way from floor to roof at 9:00 p. m. The lights over the playing floor are at this elevation and may cause this higher temperature. The heat equivalent of 57,000 watts is sufficient to raise the volume of air above the lights about 19 F in one hour or is sufficient to raise the entire volume of 3,500,000 cu ft 3.1 F. This is almost equal to the rise observed from 8:00 to 9:00 p. m.

The audience and players have a high capacity as heaters. Assuming the

average individual present is emitting 350 Btu of sensible heat per hour, this would be a total of 1,980,000 Btu. This is available for heat transmission losses and to heat up air passing in and out of the building. The heat transmission losses are estimated as 965,000 Btu at the average outside temperature of 36 F prevailing during this game. The almost 1,000,000 Btu of heat thus available will warm up about 1,736,000 cu ft of outside air to inside temperature which amounts to one-half air change per hour. The amount of steam used for space heating is not known because some of the supply is used for heating hot water and because a pipe line loss is also included. Assuming it is 1,000 lb of the 1,500 average total, another one-half air change per hour would be required to dissipate this heat. Based on one air change the velocity through the twenty 36 in. roof vents would be 410 fpm.

The absolute humidity increased 0.46 grain per cubic foot from 8:00 to 9:00 p. m. assuming that the dry bulb temperature at the humidity pick-up element under the balcony was the same as that at the 15 ft level in the center of the floor. On several occasions this has been checked and found to be very nearly correct. This increase in moisture was due to that given off by the occupants. During this period the air entered with a humidity of 2.1 gr and left with 3.91 gr at the beginning and with 4.37 gr at the end of the period. The 5,650 people would be adding 808 lb of moisture per hour to the air, based on an amount of 1,000 grains being given off per hour by each person. The following equation balances the moisture added by the audience against the moisture increase in the room volume and that carried away in the ventilating air.

$$808 = \frac{3{,}500{,}000 \times 0.46 + 2.04 \text{ (cu ft per hour)}}{7{,}000}$$

This indicates an air movement of 1,983,000 cu ft per hour which is an air change of 0.57. Any moisture evaporated from the dirt floor which is sprinkled at times would indicate a greater air change than the above 0.57. If the air change is one per hour, the moisture evaporated would be 442 lb.

At no time during any game has fogging appeared over the playing floor. Conditions seemed not far from causing this at 7:30 p. m. when the dry bulb was 54.5 F and the dew point 49 F. This is a good reason for using ventilation. At times some condensation has occurred on the roof ventilators and on the skylights. The roof insulation has prevented condensation on its under surface under all conditions encountered.

SYSTEM OPERATED RECIRCULATING

Fig. 5 shows the operation of the fan system recirculating the building air during the first one hour and twenty minutes of observation. From 7:30 to 8:00 p. m., a general drop in temperatures is shown due to the doors being open as the crowd arrived. Heat is being added but at an insufficient rate to maintain temperatures. At the start of this period, there seems little doubt but that the balcony thermostats were above 65 F so that the outlet heaters were closed. These probably came on intermittently during this half hour period due to cool air currents from the open doors. The steam consumption curve shows a considerable peak during this period.

From 8:00 to 8:50 p. m., the temperature above the 15 ft level changed but

little with a rise of about one-half degree. The rise 5 ft above the playing floor was from 55 F to 62 F during the same period. Since the audience is largely above this level, the rise probably is not due to this source of heat. It seems due to discharging from the outlets below the balconies air that entered the recirculating ducts at about 65 F and which received little heat from either the tempering coils or the outlet heater coils. The by-pass around the tempering coils would be open since the temperature of the recirculated air is about 55 F. The air was discharged then at somewhere near 65 F and since it was not much warmer than the air above the playing floor, it diffused at these lower levels. Had it been considerably warmer, its course would have been up through the balconies. The result was a 7 deg rise 5 ft above the playing floor in 50 min. A rise of 2 deg to 65 F was observed at the 15 ft level. This indicates a probable discharge temperature of 65 F.

The introduction of some outside air from 8:50 to 9:05 p. m. resulted in lowering the discharge temperature. The resulting drop in temperature at the 15 ft level was 2 F and at the 5 ft level was less than 1 F. This indicates a probable discharge temperature of 60 F to 61 F. The change back to entire recirculation at 9:05 p. m. resulted in the same rising temperature at the lower levels as experienced from 8:00 to 8:50 p. m. The average temperatures at balcony levels were 64.4 F, 64.7 F, and 64.2 F at 8:00, 8:30, and 9:00 p. m. respectively. The corresponding temperature variations were 2.5 F, 1.0 F, and 2.4 F.

The humidity record shows no definite change. From 8:00 to 8:50 p. m. with recirculation, the relative humidity decreased while the 15 ft level temperature increased correspondingly. The absolute humidity shows no significant change, although an increase would be expected from the large audience.

FAN OPERATION WITH OUTSIDE AIR

Fig. 6 shows the introduction of outside air with an outside temperature of 7 F average. The attendance was about 3,200. At 7:00 p. m., the temperature varied from 54.3 F at the 5 ft level to 67.8 F at the 95 ft level. The temperature at the balcony thermostat level was 57 F which shows the heating system under still air conditions was insufficient to bring the temperature up to the 65 F established for the thermostats. The starting of the fans with ¾ outside and ¼ recirculated air caused the steam consumption to increase from 2,500 to 7,500 lb per hour. Under these conditions, the air at all levels increased rapidly in temperature. This increase was about 5 F in one-half hour. At 7:20 p. m., 65 F was reached at the 25 ft level. This would indicate that the thermostats would prevent further heating at the outlet heaters. However, the steam consumption did not start a definite decline until 7:35 p. m. when the temperature at the 25 ft level was 68 F.

The temperatures directly above the playing floor started to decline at 7:30 p. m. and continued until 8:00 p. m. when the game started. This drop of 8 F is considered due to the opening of doors while the crowd entered. From 8:00 to 8:30 p. m., a rise of 7 F was observed at the 5 ft level while a general decline was noted at higher levels. It seems likely that the temperature at balcony thermostats was such as to cause the steam to be shut off from the outlet heaters. If so, the discharge temperature depends upon the resultant mixing of the outside and recirculated air. With the 8 F outside temperature

and with temperatures of 67 F at the recirculating inlets, the mixture temperature would be about 37 F. This would cause the by-pass to be closed and the tempering coils would raise this air to 55 F. The low steam consumption of 3,600 lb per hour during this period and the closeness of the temperature at the 25 ft level to 65 F would seem to indicate that the outlet heaters were not raising the temperature of this 55 F air. The fact that this air coming in at 55 F raised the temperature over the playing floor to 59 F from 52 F in this one-half hour period from 8:00 to 8:30 p. m. seems due to its being heated by the audience. Some of the spectators were on wooden bleachers between the playing floor and balcony. The continual dropping of temperatures at higher levels during this period seems to substantiate the belief that low temperature air was being discharged into the room. The audience, then, seems to account for this rise in temperature over the playing floor. It should be noted that the main body of air was dropping in temperature during this period.

This rise in temperature of air over the playing floor was seemingly checked by taking the entire air supply from outside from 8:30 to 8:50 p. m. The temperature dropped from 59 F to 55 F in 15 min. Since this outside air would be heated to 55 F in the tempering coil, just as was the partially recirculated air from 8:00 to 8:30 p. m., it should leave the outlet ducts at 55 F and cause a rise as before. The temperatures at the 25 ft or thermostat level were dropping and had reached 64 F at 8:30 when all outside air supply was started and they continued to drop. Probably then the air was leaving the outlets some of the time at considerably higher temperatures than 55 F. This would cause the air to rise and pass up through the balcony levels without influencing temperatures at the 5 and 15 ft levels over the playing floor.

The change to one-half recirculated air at 8:50 p. m. caused the air temperatures at the 5 and 15 ft levels to rise about 2 F in 15 min. All other temperatures increased about 1 F. The steam consumption increase from 9:00 to 9:15 p. m. is likely due to shower use.

The rise in average air temperature with 50 per cent recirculation from 9:00 to 9:15 p. m. and the drop with no recirculation from 8:30 to 8:45 p. m. seems to indicate a possible inadequacy of the heating capacity. The number of tempering coils turned on during this game is unfortunately not known. If only two were turned on, the capacity would be inadequate to heat all outside air to 55 F, but would still be capable of raising ½ or ¾ outside air to that temperature. This would aid the explanation of the results obtained.

The humidity curve rises and falls somewhat inversely as the temperature at the 15 ft level which indicates but little change in absolute humidity. The change seemed to be a decrease of 0.5 grain per cubic foot from 8:00 to 9:00 p. m.

The temperatures for the lower balcony and for the elevations the upper balcony will occupy when finished, averaged 66.4 F, 64.6 F, and 61.9 F at 8:00, 8:30, and 9:00 p. m. respectively. The corresponding variations from top to bottom of balconies was 4.5 F, 2.5 F, and 4.4 F.

Fig. 7 shows observations when taking in outside air in milder weather than that of Fig. 6. The attendance was 4,000, about 800 more than that for Fig. 6.

The relative humidity was almost 50 per cent at 7:45 p. m. The taking of ¾ of the air supply for the fans from outside reduced the relative humidity to about 36 per cent by 8:30 p. m. The air change through open doors as the

crowd came in from 7:45 to 8:00 p. m. aided in this moisture reduction. The reduction in absolute humidity was 0.9 grain per cubic foot in three-quarters of an hour. After 8:30 p. m. an equilibrium moisture content of 2.37 grains per cubic foot was maintained.

The temperatures remained practically uniform above the 15 ft level. The largest variation during the period of the game from 8:00 to 9:00 p. m. was 1 deg. The variation at the 15 ft level was only slightly greater than this. At the 5 ft level, a gradual increase in temperature took place in this period. Probably during this period the balcony thermostats kept the steam shut off from the outlet heaters most of the time. The air would then be discharged from the outlets at the fairly low temperature resulting from the mixing with recirculated air and from the heating in the tempering coils. From the fact that the 15 ft level temperatures dropped slightly while the 5 ft level temperatures were rising seems to indicate that the air reaching the playing floor was about 61 F. Some heat probably was picked up by this air from spectators as it passed through the wooden bleacher sections. When the damper positions were changed at 8:45 p. m. so as to take in only ¼ outside air and recirculate ¾ building air, the temperature did not rise at the 5 ft level but did rise about 1 deg in 15 min. at the 15 ft level. This appears due to the air being warmer at the points of discharge therefore rising above the 5 ft level over the playing floor area. The doors were opened at 9:00 p. m. which accounts for the checking of the rise in temperature.

General Discussion

In general, it may be said that many times the operation of the tempering units is not under thermostatic control. Very often only one of the three coils in parallel is supplied with steam. The effect is then to make the area of the other two coils serve as a by-pass area not under thermostatic control. The air then must be quite warm as it reaches the coil location if it is to be heated sufficiently to cause the thermostat to open up the dampered by-pass. The effect is then to hinder thermostatic control under many conditions. The reason for leaving off one or two of the coils much of the time is to prevent overheating when recirculating. When recirculating and the by-pass damper is wide open, the temperatures would still be much above the desirable duct temperatures as established on the tempering unit thermostats. An addition to this control, so that at the same time the by-pass damper is opened, a damper would close off the air passage through the coils, would obviate the necessity and undesirability of turning off the steam from one or two coils. Another variation for better control might be the placing of thermostatic valves on each of the three coil supply branches arranged so that they would open in succession.

Disturbance of Still Air Gradient by Fan Operation

Fig. 8 shows the results of fan operation on the still air temperature gradients on a cold day at a time when the building was not occupied. This was on the coldest day of the year. The mean temperature on this day was −15 F. On the previous day, it was −10 F and on the day before this it was 0 F. The average wind velocities on the three days were 13.1, 11.6, and 18.5 miles per

hour. At 8 a. m. the temperature was −26 F and at 12 noon, it was −14 F. The roof vents were closed during these days.

Readings of temperatures were taken at 9:50 a. m. from floor to roof under still air conditions. The temperature at the 5 ft level was 53 F; at the 15 and 25 ft levels, it was about 60 F. At the 85 ft level, it was slightly over 71 F

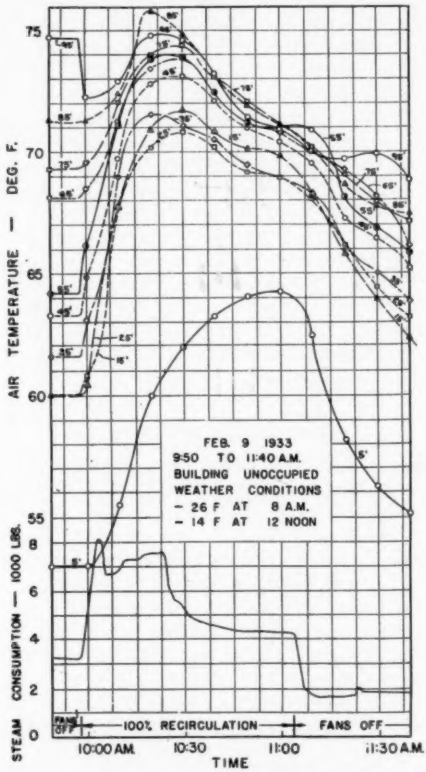


FIG. 8. DISTURBANCE OF STILL-AIR TEMPERATURE GRADIENT BY FAN OPERATION IN EXTREME COLD WEATHER

and at the 95 ft level, it was almost 75 F. This was a spread of about 22 F. Of this amount, 7 F increase occurred in the first 10 ft and almost 4 F in the last 10 ft. This will be discussed further in connection with Fig. 9.

At 9:57 and 9:59 a. m., the fans were started with 100 per cent recirculation of building air. Readings taken at 10:00 a. m. show the influence of the fan circulation. The 95 ft temperature was lowered 2½ F. The temperature at the 55 ft level was raised 2 F. Above this level, the rise was about one-half degree. Below at the 35 and 45 ft levels, it was about 1½ F. The temperature

at the 5 ft level showed no change. The readings were taken in quick succession from the bottom to the top, which influences the results somewhat since the time interval from the starting of the fans was not exactly the same for all levels. A quick response to changing temperatures on the part of the resistance thermometers is indicated and also a definite lowering of roof temperature and a raising of temperatures near the 55 ft level.

The steam demand reached a peak of 8,000 lb upon starting the fans. The previous demand had been about 3,200 lb per hour, which checks with the calculated heat loss of about 2,800,000 Btu for —15 F weather. The heavy demand for steam lasted for about 25 min. Since the recirculated air was well above 55 F, the by-pass around the tempering coils must have been open. The flow then would be divided between the tempering coils and the by-pass so that no great degree of heating would be done here. The air at the

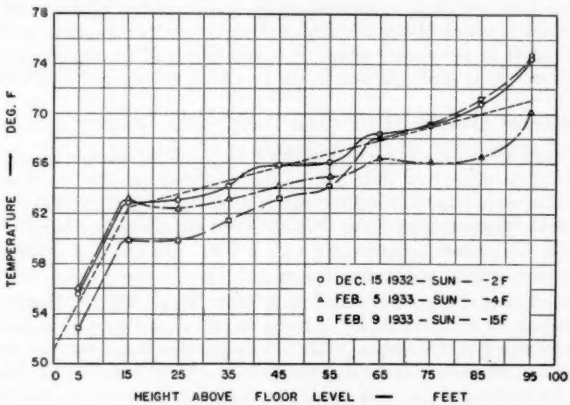


FIG. 9. TEMPERATURE GRADIENTS FOR GRAVITY CIRCULATION ON THREE COLD DAYS

thermostat level of 25 ft was at a temperature of 60 F at the outset so that the outlet heaters were responsible for considerable heating. The temperatures at all levels continued to rise until 10:30 although the 65 F temperature at the 25 ft level was reached at 10:10 a. m.

The temperature at the 5 ft level also increased, but more slowly than that at the higher levels. The temperature at this level continued to rise until the fans were shut off shortly after 11:00 a. m. The temperatures at all higher levels dropped during this half hour ending at 11:00 a. m. This was due to heat being supplied at a rate below that of the heat losses. The temperatures at the thermostat level were well above the 65 F that would permit steam to enter the outlet heaters. The main source of heat during this period would be the tempering coils. The steam consumption was about 1,000 lb per hour higher in this period than it had been previous to fan operation. An inspection at 11:00 a. m. of the thermometers on the 24 balcony thermostats showed temperatures from 67 F to 69 F with most of them at 68 F.

The spread of temperatures at 11:00 a. m. was 2.2 F from the 15 ft to the

95 ft level. The difference between the 5 and 15 ft levels was 4.7 F. The total difference was about one-third that observed before the fans were started. Since the air at the 5 ft level is below the level of the air outlets under the balcony, a low discharge temperature should be more effective in heating this level than high temperature air. Perhaps an audience seated in the balconies would aid heating at this level by retarding the upward flow of warm air. Had the balconies offered the same resistance to air flow as would obtain when filled with spectators, the difference in temperature of 4.7 F between the 5 and 15 ft levels would probably have been further decreased.

When the fans were shut off at 11:05, the difference between the highest and lowest temperatures increased. At 11:40 the total difference was almost 14 F, one-half of which was in the 10 ft between the 5 and the 15 ft levels. The curves if extended seem to indicate that still air conditions found previous to the start of fans would be obtained again shortly after 12:00 noon or about one hour after shutting down the fans. The steam consumption in this period was about 1,600 lb per hour, which is about one-half of that previous to the fan operation. This likely is due in part to heat stored in the structure.

TEMPERATURE GRADIENTS ON COLD DAYS WITH GRAVITY CIRCULATION

Temperature gradients observed under conditions of no occupancy and still air conditions on three cold days are plotted together on Fig. 9. The temperature data shown on Fig. 8 at 9:50 a. m. appears as the Feb. 9, 1933 points with −15 F outside temperature. The Feb. 5, 1933 temperatures were observed at a time when the outside temperature was −4 F. On Dec. 15, 1932, data were taken when the outside temperature was −2 F.

All three curves show an upward turn from 85 to 95 ft. Since there was bright sunshine on all three days, this was probably due to the sun effect. The rate of rise from 5 to 15 ft was considerably greater than that for the remaining distance in all three cases. The curve for −15 F weather has the steepest gradient. In each case the temperature at the 25 ft level where the thermostats are located was below that for which they were set. This shows that the steam would be on all coils and that under reverse gravity circulation the system lacked the capacity necessary to maintain conditions desired for occupancy periods. On the coldest day, this inadequacy resulted in about three degree lower temperatures at the lower levels than on the days with near zero temperatures.

The weather was not greatly different on Dec. 15 and Feb. 5. It seems impossible to account for the difference in gradient on these two days. The temperatures at lower levels were the same, whereas at the higher levels this difference became about 4 F. The previous weather does not offer an explanation. On Dec. 13, 14 and 15 the mean temperatures were 6 F, 7 F and −5 F with minimum temperatures of −5 F, −1 F and −10 F respectively. On Feb. 3, 4 and 5, they were 16 F, 2 F, and 2 F, with minimum temperatures of 12 F, −8 F and −12 F. The mean of 2 F for Feb. 5 is not a true indication of the weather at the time of the readings for that day. The weather changed rapidly in the afternoon due to warm southwest winds. The temperature was −10 F at 7:00 a. m. and −2 F at 12 noon. The average wind velocities for the three December days were 11.0, 11.4 and 9.7 miles per hour and were 8.8, 11.4 and 11.4 miles per hour for the three February days involved. The roof

vents were closed during all of these days. No explanation is known for the difference in these two gradient curves.

Since gradients under such extreme temperatures are useful for calculating the heat losses that must be met by the heating system under the severest conditions, the results for Dec. 15 are considered to show the design conditions for zero weather. A broken straight line has been plotted through the points, neglecting the sun effect. The straight line for the lower levels was intentionally lowered 1 deg that it might intersect the straight line for the higher levels at the 15 ft level.

The equation for determining the temperature at any elevation for gravity circulation from 0 to 15 ft would be

$$t = 51 + 0.76\,H$$

where

t is the air temperature at H elevation in feet
51 is the air temperature at the dirt floor.

The similar equation for temperatures at elevations from 15 ft to 100 ft would be

$$t = 51 + (0.11\,H + 9.7) \text{ or } t = 60.7 + 0.11\,H$$

where

t, H and 51 are as above and 9.7 is the difference in intercepts on the temperature axis

The equations show that the rise in temperature is about ¾ of one degree per foot in the first 15 ft and about 1/10 of one degree per foot for the remainder of the height.

The difference in temperature from the lowest point of the lower balcony to the highest point of the upper balcony is about 5 F. It will be noted that the temperatures were practically the same at the 15 and 25 ft levels. Why there was not a gradual transition from the steep gradient of the lower levels to the slower gradient of the upper levels is not known. The thermometers were checked in order to account for this behavior but no error was found. Possibly the movement of air towards the outlets which act as inlets on reversed gravity flow causes this disturbance to a smooth gradient transition. These outlets are located near the 15 ft level.

It would appear that the heating of air at the 15 ft level, the carrying of it through ducts to the 73 ft level, and there discharging it towards the peak of the building would be conducive to the maximum temperature gradient. Under these conditions of gravity flow, the fans not running, the system corresponds closely to a high outlet gravity convector heater system. The inlets are below the balconies at the 15 ft level and the main sources of heat are located here. Additional heat is added a short distance above this in the tempering coils. The discharge is at the 73 ft level through the 12 recirculating openings located at that elevation. Building air probably enters the lower recirculating inlets of the fan system located at the 58 ft level, flows upward and mixes with heated air which is discharged through the openings at the 73 ft level.

The location of the inlets of these gravity convectors 15 ft above the floor probably accounts for the abrupt change in gradient experienced at the 15 ft level. The location of the outlets at an elevation of 73 ft would appear to be conducive to high ceiling temperature.

Conclusions

The system as installed is considered to be well suited to the heating and ventilating requirements of the building. During inoccupancy periods, gravity circulation places a limit on the heating capacity and therefore the cost of heating. During occupancy, the balconies are heated by a high temperature discharge from outlets placed below them. When the desired balcony temperature is reached the steam is shut off from the outlet heaters and air is then discharged at the lower temperature desired at the playing floor and diffuses over the lower levels. When the air over the playing floor is warmer than desired, this entering air will lower the temperature. If it is cooler than desired, the entering air will raise its temperature.

Under gravity circulation and near zero outside temperature conditions, a temperature gradient of about ¾ of a degree per foot was found for the first 15 ft of elevation and about 1/10 of a degree per foot from 15 ft to 97 ft at the peak of the roof. While these results apply strictly only to a building and a heating system similar to that tested, it is considered that this gradient would be approached in any large single room building, exposed on all sides and on the ceiling and with heating units located 15 ft or less in height above the floor. Should radiant heaters be placed at higher elevations the gradient probably would be increased.

A fan system is considered desirable in this type of building in order to reduce humidity by the introduction of outside air and at times to remove the heat from the audience. The fan system when operated recirculating reduces the temperature gradient. On a −15 F day and no occupancy, the gradient of about 22 F with gravity circulation was reduced to approximately 7 F with fan recirculation. Just how much of a temperature gradient will exist under fan recirculation depends upon, among other factors, the size of the audience and the discharge temperature from the outlets.

It is considered safe and desirable in this type of a building that the designer use the temperature gradient observed under gravity circulation rather than any found with fan recirculation.

DISCUSSION

W. W. Timmis: In the opinion of the authors, are the temperature gradient and, particularly, the characteristic break at the 15-ft level due to the shape and size of the building studied and to the arrangement of the heating system, or would a curve with similar characteristics be obtained in buildings of similar type although not necessarily with the same measurements or the same type of heat distribution system?

Authors' Closure: The temperature gradient curve no doubt has a definite relation to the building proportions, type of construction and to the heating system. The characteristic break at the 15 ft level shown in Fig. 9 for gravity circulation is caused at least in part by the location of the fan system outlets, which act as inlets on gravity circulation. The location of these outlets is shown on Fig. 2. There would seem to be no active circulation at these times below the level of these openings.

The location of the recirculating openings about 65 ft above the floor level would tend to make the temperature gradient steeper in this building than in many. The measured temperature gradient is less steep than formulas used hitherto would indicate.

No. 959

INDICES OF AIR CHANGE AND AIR DISTRIBUTION

By F. C. Houghten † and J. L. Blackshaw ‡ (*MEMBERS*)
Pittsburgh, Pa.

VENTILATION engineers have for a number of years accepted the principle of analyzing air for carbon dioxide concentrations as a measure of the rate of air change, and of comparing these concentrations to measure the uniformity of air distribution throughout a ventilated space. Although investigators in carrying on original research have employed the CO_2 method to some extent, it has found little use by engineers in making routine ventilation surveys in schools and audience halls because of the difficulties involved in obtaining representative samples of the air in a room, in preserving these samples during their transportation to a laboratory for subsequent analysis, and in the extremely precise laboratory determination that is necessary to measure with sufficient accuracy the small quantity of CO_2 in the sample.

Recently, there was written into the *Code of Ventilation Standards* of the American Society of Heating and Ventilating Engineers a section approving the use of the CO_2 method as a measure of air distribution. Questions arose concerning the desirability of having, in a Code to be universally applied, a standard which is so little used. As a result, it was suggested that the Research Laboratory make an analysis and comparison of the CO_2 method and other methods of measuring air change and air distribution.

An occupant of an enclosed space changes the physical and chemical composition of the air contained therein in several ways. Any of these changes might be used as a measure of ventilation, but the three most adaptable are: the increase in the sensible heat content of the air; the increase in the moisture content of the air and therefore its latent heat content; and the increase in the CO_2 content of the air.

Air Change Indices

Data [1] published by the Research Laboratory of the American Society of Heating and Ventilating Engineers, as well as results of other investiga-

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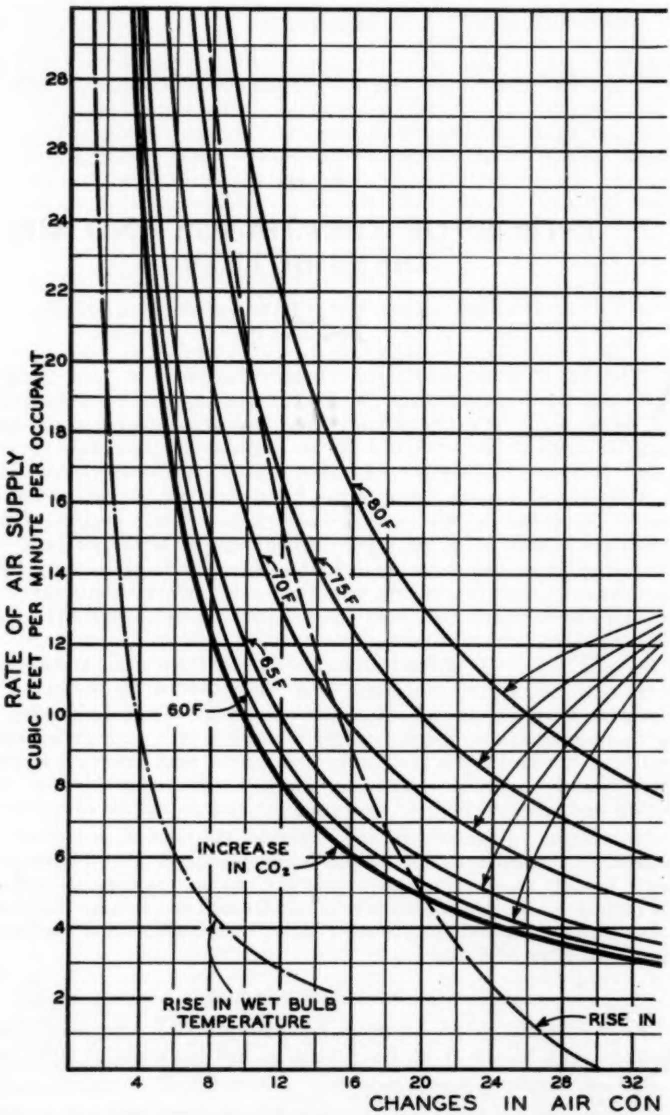
[1] Heat and Moisture Losses from the Human Body and Their Relation to Air Conditioning Problems, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant. A.S.H.V.E. Transactions, Vol. 35, 1929, p. 245.

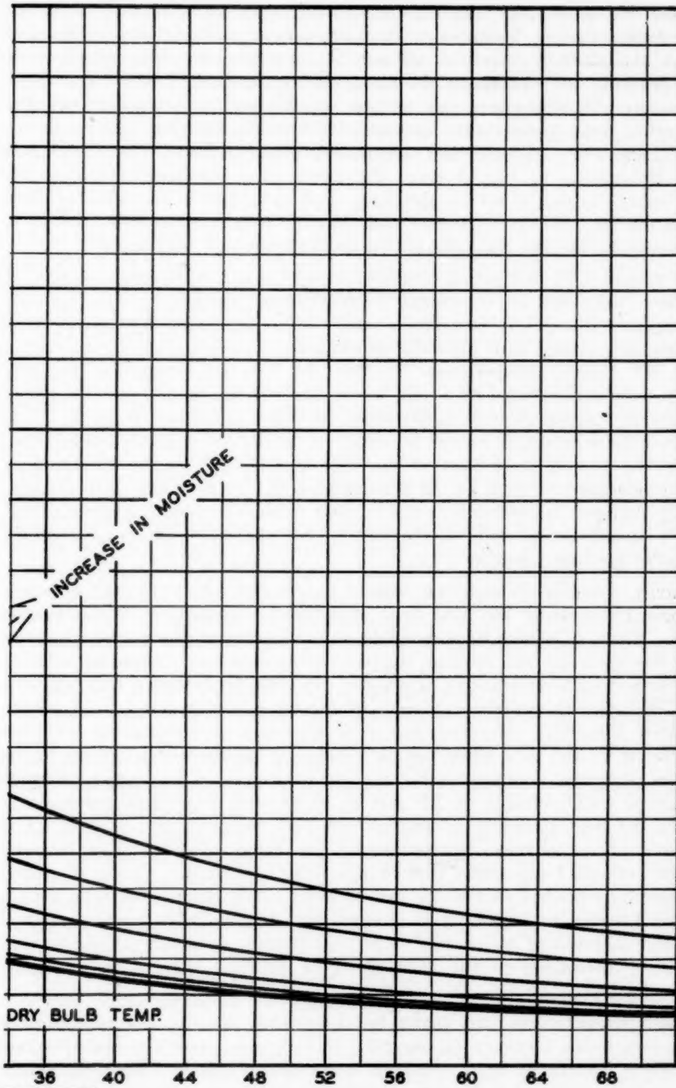
Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Hotel Statler, Detroit, Mich., June, 1933, by F. C. Houghten.



CHANGES IN CO_2 CONCENTRATION IN PARTS PER 10,000 AIR. CHANGES IN MOISTURE

Change of CO_2 content applies to any atmospheric condition over a large temperature range. Change in moisture content is for any atmospheric condition having the dry bulb temperature indicated.

FIG. 1. RELATION BETWEEN CHANGE IN AIR



DITIONS

CONTENT IN GRAINS PER POUND OF DRY AIR. CHANGES IN TEMPERATURE IN DEGREES F

Change in wet-bulb and dry-bulb temperatures from initial condition of 70 F dry-bulb and 40 per cent relative humidity. Volume of air supply corrected to 29.92 in. barometric pressure, 70 F dry-bulb, and 40 per cent relative humidity.

CONDITIONS AND AIR SUPPLY PER OCCUPANT

tions, show that an average size man seated at rest in an audience hall, under ordinary temperatures, dissipates to the atmosphere 0.59 cu ft of CO_2 per hour, and also definite quantities of sensible heat and moisture which vary with the dry-bulb temperature of the air according to established relationships. Based on this CO_2 dissipation rate and on sensible and latent heat dissipation rates varying with temperature in accordance with accepted curves[2] from Research Laboratory data on heat and moisture losses from men at rest and at work, the curves in Fig. 1 show the effect of a man on the air in an enclosed space, assuming all of the CO_2, moisture, and heat added by him remain in the air. An average size man, having a surface area of 19.5 sq ft as determined by the Du Bois chart [1, 3] is used throughout this paper.

As the rate of CO_2 dissipation remains the same for a wide range of normal temperatures only one curve results from plotting changes in CO_2 concentrations against changes in ventilation rates. Since moisture dissipation from the human body varies with dry-bulb temperature, a series of moisture curves is given for various temperatures so that the rate of ventilation for any temperature between 60 and 80 F can be obtained by interpolation. The curve for change in sensible heat is based upon an initial condition of 70 F. The use of dry-bulb temperature rise as an index of ventilation is precluded because, as verified by test, a large part of the heat is absorbed by the surrounding walls and furniture as soon as the temperature of the air rises. Of the two remaining changes, concentration of CO_2 and moisture content, assuming that neither is absorbed by walls or furniture, the one more easily determined should make the better index.

Ordinarily, increase in moisture content is measured by a change not only in wet-bulb temperature but also by a simultaneous change in dry-bulb temperature. Since it would be impossible to predetermine these two changes, a curve has been plotted on Fig. 1, *for comparison only*, showing relative changes in wet-bulb temperature when the dry-bulb temperature remains constant at 70 F and the moisture content of the air varies upward from 40 per cent relative humidity. Reading this wet-bulb curve in comparison with the CO_2 curve, it is seen that when the air change is approximately 10 cu ft per minute per occupant, a change in the rate of air supply of 1 cfm per occupant will result in a CO_2 change of 1.0 part in 10,000 and a wet-bulb temperature change of 0.4 deg, assuming a constant dry-bulb temperature of 70 F. For other rates of air change, approximately the same relation holds between change in wet-bulb temperature and change in CO_2 content. Hence, if a wet-bulb temperature change of 0.4 deg can be more easily determined than a CO_2 change of 1.0 part in 10,000, it is obvious that the moisture content method is the better measure of air change or air distribution, assuming of course that the moisture dissipated to the atmosphere is retained therein to the same extent as the CO_2.

Wet-bulb temperatures can easily be determined by the use of ordinary thermometers to an accuracy of one-half of a degree; with the use of better thermometers, accurately read, wet-bulb temperatures can be determined to an accuracy of one-tenth of a degree. Compared with these simple readings, the

[2] Heat and Moisture Losses from Men at Work and Application to Air Conditioning Problems, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant (Figs. 2 and 3). A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 541.
[3] Du Bois, D. and E. F. (*Archives of Int. Med.*, 1916, Vol. 17, p. 865).

determination of CO_2 content in the air is much more difficult, because it involves the collection of samples of air, the preserving of them while they are being transported to the laboratory, and the use of highly specialized apparatus, such as the Peterson-Palmquist analyzer, which requires infinitely greater skill and manipulation than that needed in the taking of sling psychrometer readings. This reasoning would indicate that the moisture content of the air is a more convenient and usable measure of air change in a ventilated space than is its CO_2 content.

TEST METHODS

In order to verify the above deductions and in order to determine the extent to which either the moisture or the CO_2 dissipated to the atmosphere by an occupant may be absorbed by surrounding objects, a number of tests were made in the psychrometric chambers of the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS at the United States Bureau of Mines Experiment Station, Pittsburgh, Pa. Fig. 2 is a sketch of the two rooms which are more fully described in a previous Laboratory report.[4] The temperature and moisture content of the air in the second chamber were held constant both during a test and for a considerable time before it by ventilating the room with a large volume of air at fixed temperature and dewpoint, supplied from the air conditioning equipment located in an adjoining room. The first chamber, in which the tests were made, was independently ventilated with air from the second chamber. When the conditions in both the first and second chamber had remained constant and equal to the conditions supplied by the air conditioning equipment for a sufficient length of time to insure equilibrium, the test was begun by admitting the subjects and seating them at rest uniformly over the floor area of the test chamber. It was assumed that any changes in the properties of the air in the test chamber during the test period resulted from occupancy. From eight to ten adult male subjects participated in the respective tests. Their rate of CO_2 production and heat and moisture dissipation was calculated from their body surface areas as given by the Du Bois Chart and the Laboratory data for the air conditions pertaining in the test room.

Difficulty was experienced in controlling the amount of ventilation in the first chamber to a predetermined quantity of air. By one method, air was moved from the second chamber to the first by building up a pressure in the second chamber, as a result of which air passed into the first chamber from which it was allowed to escape through a calibrated orifice into the corridors of the building. An attempt was made to measure the quantity of ventilation by these orifice readings, but it was found that exfiltration through the walls of the chamber, even with the small pressure differences maintained, amounted to more than the air flow through the orifice. Another method maintained equal pressures, as measured on a Wahlen gage, in both chambers and in the corridors and allowed no movement of air to or from the first chamber other than that through a two-foot square opening in the doorway between the first and second chambers, through which the air was allowed to circulate at will between the two as through an open window. This method gave uniform conditions throughout a test, but it did not give an accurate

[4] Determining Lines of Equal Comfort, by F. C. Houghten and C. P. Yaglou. A.S.H.V.E. TRANSACTIONS, Vol. 29, 1923, pp. 163-175.

measure of the quantity of air used to ventilate the test room, and hence, while it gave a satisfactory comparison between the CO_2 and the moisture content methods of determining ventilation, it did not give an absolute check on either. The most successful test conditions were had by using a small blower to move air from the second chamber into the first through a calibrated orifice, and by controlling the size of the exit opening from the first chamber into the corridor of the building, so as to maintain a pressure of approximately 0.001 of an inch

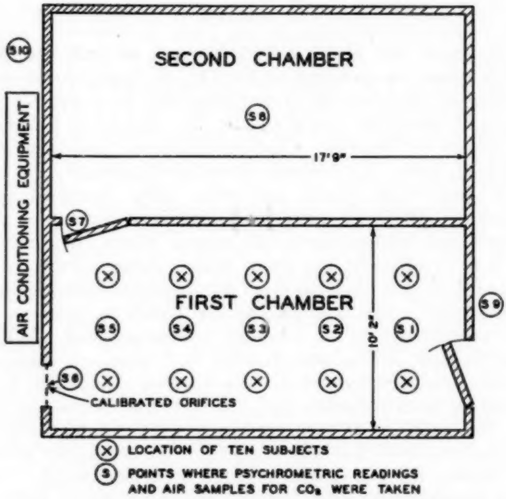


FIG. 2. PSYCHROMETRIC CHAMBERS USED IN VENTILATION STUDY

of water column in the first chamber over that in the corridor and in the second chamber. This method was used in most of the tests. The methods failing to give control are mentioned to show that large leakage losses usually result in ventilated space, in addition to the ventilation load supplied under control by the installed equipment.

A number of tests were made, each with a different rate of ventilation as measured in cubic feet per minute per occupant of air corrected to 29.92 in. barometric pressure, 70 F dry-bulb, 40 per cent relative humidity. During the progress of each test samples of air for analysis for CO_2 were taken and readings of the sling psychrometer were made at various stations within the test room, in the second chamber, and in the corridors. To prevent contamination, the gas samples were held in clean, dry, mercury sealed bottles. To avoid discrepancies in temperatures, all wet-bulb and dry-bulb readings were taken on the same sling psychrometer. Each test was continued until constant conditions in the first chamber had been reached and maintained for a sufficient length of time to insure equilibrium. The average time taken for each test was ninety minutes. From the increase in CO_2 concentration between the initial

and final conditions as indicated by the analyses of the samples collected, the air change in cubic feet per minute per occupant was calculated from the following formula which applies after equilibrium has been reached:

$$R = \frac{\frac{C}{60} \times \frac{S}{19.5}}{P \times 10^{-4}}$$

where:

$R =$ ventilation rate in cubic feet per minute per occupant (standard conditions).
$C =$ CO_2 production rate, in cubic feet per hour per average sized man of 19.5 sq ft (standard conditions).
$S =$ area of average subject, in square feet.
$P =$ increase in CO_2 concentration, in parts per 10,000.

Similarly, the air change was calculated from sling psychrometer readings applied to the psychrometric chart as follows:

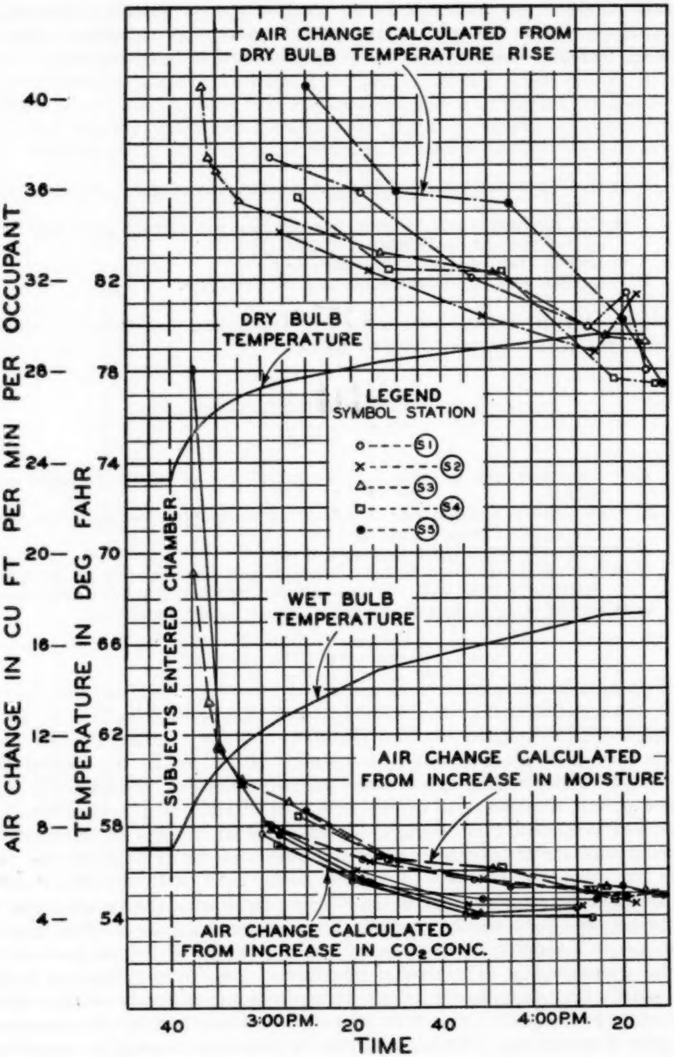
$$R = \frac{M \times \frac{S}{19.5}}{G \times w}$$

where:

$R =$ ventilation rate in cubic feet per minute per occupant (standard conditions).
$M =$ moisture production rate in grains per minute per average sized man of 19.5 sq ft (standard conditions).
$S =$ area of average subject, in square feet.
$G =$ increase in moisture content in grains per pound of dry air as actually determined from wet- and dry-bulb readings and psychrometric chart.
$w =$ weight of air in pounds per cubic foot (standard conditions).

TEST RESULTS

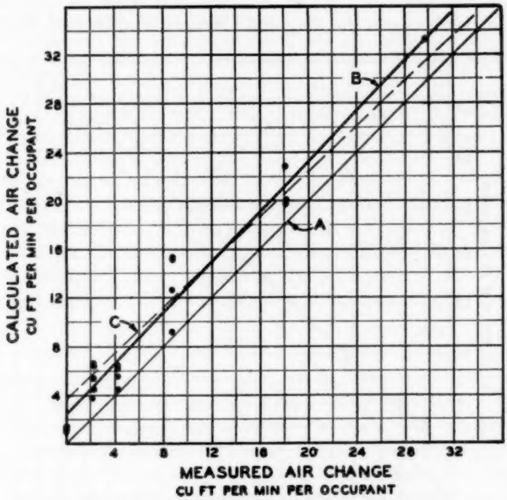
Air changes in a typical test as calculated from CO_2 and moisture increase at five respective sampling stations are plotted against time in Fig. 3. In this test, ten subjects occupied the first chamber, and the ventilation both inward and outward took place through an opening 10 in. by 29 in., located 8 in. above the floor in the doorway between the two chambers. Because this set-up did not permit measurement of the actual ventilation, the results could not be checked to a known air change, but they are of interest in indicating the uniformity of air distribution to the different stations throughout the room. While the volumetric capacity of the test room, 1450 cu ft, required considerable time for the establishment of equilibrium conditions, the curves show that equilibrium was very nearly reached 70 min after occupancy. This test was of interest in a number of respects. First, the results of the analyses of CO_2 and the determination of grains of moisture at the different stations indicate very satisfactory distribution of air throughout the different stations in the room in spite of the use of a type of ventilation which would be expected to give poor distribution. This air distribution was later checked by smoke tests which showed that convection currents in the neighborhood of each occupant, set up by heat dissipation from his body, mixed the air in the room. Second, the indicated air changes as calculated from CO_2 increases and from grains of moisture increases differed little. A slightly greater air change is indicated by the grains of moisture method in this particular test, but this relation was not



Relation between time and air changes as calculated from CO_2 increase, moisture increase and dry-bulb temperature rise. Also the relation between time and dry-bulb and wet-bulb temperatures of the test room.

FIG. 3. DATA FROM TYPICAL TEST

uniformly substantiated in later ones. Third, the test shows a surprisingly high air change for the room, considering the small opening through which



Data points and Curve B from CO_2 analysis. Curve C from moisture determinations, Fig. 5. Curve A, measured air change.

Fig. 4. Relation between Measured Air Change and Air Change Calculated from Air Conditions

both the incoming and outgoing air had to pass, with no pressure difference or wind effect to cause circulation.

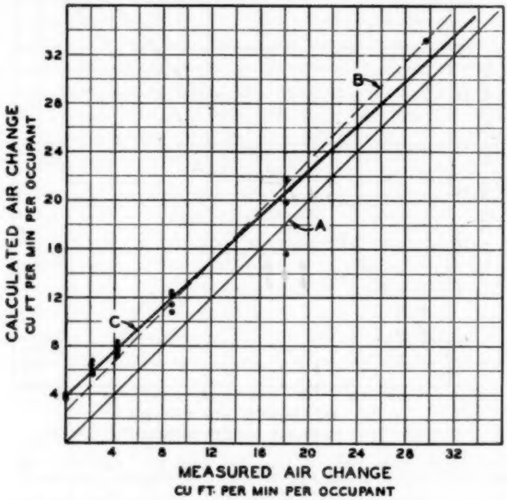
The curves, "air change calculated from dry-bulb temperature rise," shown in the upper part of Fig. 3 were obtained by calculating air change from differences in sensible heat as determined by dry-bulb temperature rises. These curves indicate ventilation rates about eight times as great as those found by calculations using CO_2 or moisture content, and are so irregular and inconsistent with the estimated ventilation in the room, that they show that sensible heat is not a satisfactory measure of air change. The rises in both wet and dry-bulb temperature resulting from occupancy are also plotted.

The results of CO_2 analysis and grains of moisture determination for all tests are plotted in Figs. 4 and 5, respectively, where the calculated ventilation in cubic feet per minute per occupant is plotted against the known ventilation in cubic feet per minute per occupant as measured when supplied by the ventilating system. The results obtained by both methods of calculation are higher than the measured air change. With the exception of one point, at an air change of 18.1 cu ft, the results of grains of moisture determinations are much more consistent and show less variation between individual points than do those of the determinations resulting from CO_2 analysis.

For comparison, Curve B, plotted in Fig. 4, is shown as a broken line on Fig. 5, and Curve C, plotted in Fig. 5, is shown as a broken line on Fig. 4.

It may be seen that there is a close relation between the CO_2 and the moisture curves, and that they bear similar relationships to the measured ventilation curve.

The discrepancies between the diagonal Curve *A* and Curves *B* and *C* in Figs.



Data points and Curve *C* from moisture determinations. Curve *B* from CO_2 analysis, Fig. 4. Curve *A*, measured air change.

FIG. 5. RELATION BETWEEN MEASURED AIR CHANGE AND AIR CHANGE CALCULATED FROM AIR CONDITIONS

4 and 5 which show larger calculated air changes than those measured by the ventilating system may be accounted for in several possible ways: (1) the dilution effect of the air represented by the original volumetric capacity of the room over the ventilation rate during the test period; (2) the absorption of part of the CO_2 and water vapor dissipated to the atmosphere by the walls and other surfaces within the room; (3) possible error in the measurement of the air supplied; (4) unmeasured infiltration; (5) possible error in assumption that the rates of CO_2 production and moisture dissipation by the occupants are the same as those for the conditions of the tests reported in the earlier laboratory studies.

DILUTION

The effect of the volumetric capacity of the room on the concentration of CO_2 and water vapor in it at the end of the test period may be calculated by the dilution formula:[5]

$$Q = \frac{V_r}{V_v} V_{gas} \left(1 - e^{-\frac{V_v}{V_r} t}\right) + Q_0 e^{-\frac{V_v}{V_r} t}$$

[5] Carbon Monoxide Distribution in Relation to the Ventilation of an Underground Ramp Garage, by F. C. Houghten and Paul McDermott. A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 439.

where:

Q = volume in cubic feet of carbon dioxide or water vapor in the room at any time, t.
t = elapsed time in minutes.
V_r = volume of room in cubic feet.
V_v = volume of air supplied in cubic feet per minute.
V_{gas} = volume, in cubic feet, of carbon dioxide or water vapor added to the room per minute.
Q_0 = volume in cubic feet of carbon dioxide or water vapor in the room at the beginning of the time considered, or when $t = 0$.
e = base of the Naperian system of logarithms, 2.71828.

Except for very low air change rates, application of the above formula shows that there was little error introduced by dilution in the test chamber due to the original volume of air in the room. When the test chamber was occupied for 90 min by ten subjects and was ventilated at a constant rate of 10 cfm per occupant, the error introduced in using equilibrium conditions in the calculations amounted to less than one per cent.

Absorption

No satisfactory measure can be made of the effect of absorption of water vapor and CO_2 from the air by walls and other surfaces. It is generally assumed that under changing temperature and humidity conditions such surfaces have a high affinity for water vapor but not for other gases. There are no data available, however, to prove that water vapor in the amounts in which it is contained in the atmosphere under the conditions of these tests should act differently than CO_2 or any other gas. In fact, one may assume that with changes in temperature and CO_2 concentration, plaster and other masonry material, largely made up of carbonates, might absorb or expel relatively more CO_2 than water vapor. It would seem expedient to accept the few data presented in Figs. 4 and 5 as evidence that absorption is not a considerable factor in either method. It must, however, be borne in mind that when glass or other surfaces in a room are cooled below the dewpoint of the atmosphere so that condensation takes place on them, the grains of moisture determination method could not be used to measure air distribution. Also, if the amount of ventilation in a room is so low that occupants cause the air to approach saturation, the water vapor method could not be used with confidence. Further, in cases where the air in a room is humidified, care must be taken to determine the moisture content at the source of the air with which the room is ventilated. This would not be the moisture content of the air outside, but that of the air entering the room from the conditioning system.

Error in Assumptions

Erroneous assumption of the rates of CO_2 production and moisture dissipation by the test subjects may easily be a factor affecting the discrepancy between the diagonal Curve *A* and Curves *B* and *C* in Figs. 4 and 5, by reason of a difference of degree of activity of the subjects in the original Laboratory studies of heat and moisture dissipation and of the subjects in the present tests. An attempt was made to keep the subjects in both studies seated at rest as they would be in a school or audience hall. However, the short duration of the present series of tests compared with the longer duration of the series

in the earlier study resulted in less effective discipline of the subjects in the present study. It should be noted that increased physical activity of the subjects increased the rates of dissipation of both CO_2 and moisture by approximately the same percentages, and that such increase in activity would help account for the discrepancy between Curve *A* and Curves *B* and *C*.

The increase in CO_2 or grains of moisture content of the air in a ventilated space may be used for two different purposes in a ventilation survey. First and more usually, it may be used as a measure of uniform air distribution throughout the room; second, it may be used as a measure of the total volume of air being supplied to the room. If used for the first purpose, errors in rate of CO_2 or rate of moisture dissipation will have little effect, for the interest lies in the variation of distribution of air through the room. If, however, either method is used to determine the total quantity of air being supplied to the room, the accuracy of the result will depend on the accuracy of the assumption of rate of dissipation of CO_2 or water vapor. The two Laboratory papers [1, 2] dealing with heat and moisture losses from the human body and giving results for a person seated at rest as in an audience hall and for a person working at diverse fixed rates, and data gleaned from reports of other investigators on variation of metabolism with variation in activity, furnish bases for a fairly satisfactory assumption of rates of CO_2 and moisture dissipation from occupants in a ventilated space.

ODORS

While the study was not designed as a comprehensive investigation of the relation of odors to air change, it did offer an opportunity for a few observations which may be of interest. For all tests made with air changes greater than 10 cfm per occupant the air in the room seemed free from odors or depressing effects. For air changes of 10 cfm per occupant odors were noticeable to a person entering the room from the outside atmosphere, and the general condition of the room air seemed depressing and stuffy. However, in all cases where the air change was 10 cu ft or less the temperature was uncomfortably high. The tests were usually started at approximately 70 F dry-bulb, but the heat dissipation from the subjects made it rise about 6 deg when the air change was 10 cfm per occupant, and more when the air change was less than 10 cu ft. Since it is generally accepted that odors are closely related to temperature, increasing rapidly with its rise above 70 F, the odor effects noticed with air changes of 10 cu ft and less may have resulted from the high temperatures rather than from the small air changes. For air changes of 5 cfm per occupant or less, the condition in the test room became distinctly obnoxious and depressing to a person entering from the outside. With ten subjects, the test room had a capacity per occupant of 145 cu ft which made it somewhat more crowded than is accepted as good practice in schools.

AIR MOTION IN RELATION TO AIR DISTRIBUTION

The inconsistency shown on Figs. 4 and 5 between the points determined from CO_2 analysis and those obtained from the moisture content determinations may be caused by differences in physical functions and technical methods. The human body dissipates CO_2 to the atmosphere by the exhaled breath, which is

ejected with concentrations of CO_2 ranging from 200 to 500 parts in 10,000 parts of air. Hence, large variations in concentrations of CO_2 may be found at points in the atmosphere two or three feet from the occupant for small periods of time even with what might otherwise be considered good air distribution. Since the samples of CO_2 are customarily taken by hand, it is reasonable to suppose that even with the utmost care, the observer's own breath may sometimes contaminate them.

At a dry-bulb temperature of 70 F, about 35 per cent of the moisture dissipated to the atmosphere by the human body comes from the respiratory tract. The remaining 65 per cent comes from the body surface. Thus, since approximately one-third of the possible water vapor dissipation comes from the breath, there is less likelihood than with CO_2 of having large concentrations of water vapor at points in the atmosphere a few feet from the occupant in a well ventilated room. Further, the very nature of the method of making a sling psychrometer reading tends to break up any concentrated pockets of water vapor and to give an integrated reading of the moisture content of a circle from 30 to 40 in. in diameter about 2 ft in front of the observer.

It is probable that these differences in a large measure account for the fact that CO_2 points in Fig. 4 are less consistent than the grains of moisture points in Fig. 5. It seems, however, that the difference in measurement and dissipation of moisture should not affect the true purpose of the determination. A room in which air is well distributed and where air motion is sufficient for good ventilation might conceivably give samples of air in which a breath of high concentration CO_2 has floated. The grains of moisture determination, because obtained by an integrated reading over a fairly large area in any part of a room, would seem to be representative of the air supply to that part of the room.

Consideration of this question suggests that air motion throughout a room might be a better index of air distribution than either CO_2 or grains of moisture. In none of the tests here recorded, even with highly localized points of air admission and exit, was there any consistent variation shown in the distribution of CO_2 or water vapor as between the end of the room where the air was supplied and removed and its opposite end. However, it should be borne in mind that this room was small. Convection currents seemed to be quite effective in mixing the air. Perhaps the distribution of air to each particular corner or section of a room by the ventilating system is not so much needed in a well ventilated space as is sufficient air motion throughout a room to break up stagnant air pockets next to occupants. It is accepted in ventilating practice that for good ventilation, the air in all parts of a room should have a velocity of from 20 to 50 fpm as measured by the Kata thermometer. If air could be supplied to every section of a room without air motion, the ventilation would still be unsatisfactory, but if such air velocities do exist locally throughout a room, good mixing or uniform distribution will be had whatever the location of the supply and exhaust. Hence, the question arises as to whether or not air velocities as measured by the Kata thermometer throughout a room might not give better assurance of mixing and air distribution than is given by either CO_2 or water vapor. It would seem that this question could bear further consideration. Obviously, air motion cannot be used as a measure of the rate of air supply.

Application

The curves presented in Fig. 1 may be used directly for solving air change and air distribution problems. For example, the characteristics of air supplied to ventilate a room are:

CO_2 content	4 parts per 10,000
wet-bulb	55.0 F
dry-bulb	68.0 F
moisture content (from psychrometric chart)	43.0 grains per pound of dry air

and the characteristics of the air in the ventilated, occupied room under equilibrium conditions are:

CO_2 content	10 parts per 10,000
wet-bulb	59.1 F
dry-bulb	72.0 F
moisture content (from psychrometric chart)	53.3 grains per pound of dry air

The increase in CO_2 is 6 parts in 10,000, which from the CO_2 curve in Fig. 1 gives an air change of 16.5 cfm per occupant. The increase in grains of moisture is 10.3 grains, which from an interpolation of the moisture curves in Fig. 1, shows that at a temperature of 72.0 F, the air change is also 16.5 cfm per occupant.

Summary and Conclusion

Calculations determined from CO_2 and grains of moisture increases indicate a somewhat higher air change than that supplied by the ventilating system. The difference may result from absorption of CO_2 and water vapor by walls and other surfaces or from erroneous assumptions of the rates of CO_2 and moisture dissipation as taken from the earlier Laboratory studies. However, the discrepancies noted are not sufficient to seriously affect the use of either method as a measure of air change or air distribution.

Greater consistency is shown among individual determinations of air change made by the moisture content method than among determinations made by the CO_2 method.

Measurement of air change in a room by sling psychrometer can be made more quickly and with much greater ease than by the lengthy and complicated CO_2 method. Since the accuracy of the wet- and dry-bulb method is as great or greater than that obtained through the use of CO_2, it is recommended as the more desirable method for making ventilation determinations.

Observations made in a room with controlled and measured rates of air change indicate pronounced odors for air changes less than 10 cfm per occupant when the dry-bulb temperature was above 75 F.

The tests and the deductions suggest an air motion of from 20 to 50 fpm, determined by the Kata thermometer, as a possible index of satisfactory air distribution.

DISCUSSION

A. P. Kratz: In connection with this paper, the experience at the Research Residence at the University of Illinois, under operating conditions in a full-scale building, may be of interest. For winter operation, various types of water pans were used with a gravity warm air furnace, and a number of attempts were made to calculate the air infiltration from the relative humidities and the water evaporated. No domestic operations were carried on and the only apparent source of evaporation was from the water pans. In every case the infiltration calculated in this way was much too low, ranging from 0.1 to 0.5 air changes per hour. Calculations from heat losses and fuel burned indicate approximately one air change per hour, which seems reasonable for the character of construction. Apparently water vapor was coming from some source outside of the water pans. The most probable source was the basement walls and floors.

Tests during winter operation with an air washer, in which much greater quantities of water were evaporated than in the case of the water pans, indicated that, at outdoor temperatures lower than 5 F, the water evaporated was greater than that required to maintain the observed relative humidity of 35 per cent, and at outdoor temperatures higher than 5 F it was less than the theoretical amount. Thus it was evident that during winter operation there was an interchange of water vapor between the air indoors and the materials in the structure, and the weighed amounts of water evaporated were not representative of the amounts appearing in the indoor air. In the case of summer operation, calculations made during the cooling tests, using the observed relative humidities and the condensation collected from the surfaces of the cooling coils, resulted in from 0.75 to 1.0 air changes per hour. This appeared reasonably accurate for summer conditions.

Hence, the method proposed for using the water vapor as an index, while undoubtedly accurate under conditions where the interchange of water vapor between the air and the structure is negligible, may prove to be reasonably accurate at times in a building under actual operating conditions, and may result in entirely misleading indications at other times.

J. N. Hadjisky: The determination of air change and distribution in an ordinary large room by means of measuring the change of absolute moisture content of the room does not seem to be a reliable method. I have observed in tests made during the last year that, if a room is kept at a lower relative humidity than the outside, there is a noticeable emission of moisture from the wall surfaces. This emission or diffusion inward begins to be noticeable when there is a difference of 10 or 20 per cent in the relative humidity, with the dry-bulb temperature remaining nearly the same.

When the process is reversed and steam is injected in the room to increase the humidity, the walls will absorb and diffuse considerable quantities during a period of 5-6 hours.

While measurement of moisture content is easier than the CO_2 content determination, it is necessary to determine first the rate at which moisture is given off or absorbed by the wall and other surfaces for a given difference of vapor tension or relative humidity between the outside and inside, so that a proper correction factor may be added.

My observations were made in a test room, the wall construction of which was ½-in. beaded ceiling, 3-in. insulating board and 1-in. ship lap on the outside. The outside was painted with 2 coats of green paint and the inside surfaces were given a primary coat of cream colored paint and 2 coats of varnish. The cement floor was painted with 2 coats of waterproof paint. The room was 12 x 18 ft x 9 ft ceiling.

The moisture emission was about 4 lb in 3 hours, by condensation, and the absorption was about 1½ lb per hour for a period of 5 hours. This test room was constructed in a large factory building.

From the facts mentioned, it appears that in summertime a room, if conditioned to a lower moisture content and relative humidity than the outdoors, will give a very high rate of air change by infiltration, if measured by the change in absolute moisture content.

In winter, the same room will tend to show a lower rate of air infiltration due to the higher vapor pressure inside the room and the tendency of the moisture to diffuse outward.

E. K. CAMPBELL: Although the information contained in this paper is not conclusive in some respects, it is an important step in the right direction. I have felt that a weakness in the Society's code of ventilation requirements lay in the definite requirement of the carbon dioxide test for distribution. I think there are not a dozen engineers in the United States who could make an accurate carbon dioxide test. I am sure there are none west of the Mississippi River, and there is quite a territory out there where some tests might be desired.

It is not as important for practical purposes that a field test shall be entirely and scientifically accurate as it is that it shall be used. Laboratory tests are usable to set a standard, but a field test should be understandable to the ordinary engineer and school superintendent, so that they can make a simple test which will be sufficiently accurate for practical purposes. The sling psychrometer seems to meet such a need.

I hope this work will be continued until some definite conclusion can be reached, and a reasonable field test included in our code of ventilation requirements.

No. 960

TESTING AND RATING OF AIR CLEANING DEVICES USED FOR GENERAL VENTILATION WORK

By Samuel R. Lewis † (*MEMBER*), Chicago, Ill.

FOR years there has been widespread discussion and controversy on the subject of how air cleaning devices should be tested and rated. No matter how this testing was done, however, an efficiency of dust removal of better than 97 per cent has nearly always been claimed in the literature of manufacturers, whether the air cleaning device was the crudest old-time water spray or the most refined modern fabric filter or viscous coated impact surface. This high claim for arrestance of dust was safe enough with some methods of testing because no two people could agree on how to prove or disprove it.

Who, of the ordinary busy world, not especially interested in the fabrication of air cleaning devices, has given much thought to this question of measuring dust removing efficiency? Who has considered particularly whether the advertised efficiency represented a new clean device or whether it represented the conditions at some average time during the period of operation between clean and necessary-to-be-cleaned? Who knew or worried whether the efficiency was determined by laboratory test or by field test?

Anyway, how could the air cleaning efficiency be protected by a shot gun guarantee if one realized that no two dusts are alike? There is a different kind of dust, for instance, in Chicago from that in New York, and the dust in St. Louis isn't like either of these, nor like Pittsburgh dust. The dust in the Chicago loop is different in concentration and type from that on the Midway, and an air cleaning device on the 20th story taking air through a window has a very different problem from the air cleaning device in the sub-basement of the same building, taking its air from grade level in the alley.

The soot from bituminous coal and oil combustion, the most destructive factor in modern ventilation work, defeats very easily the 98 per cent efficiency hopes of the air washer which uses the imperfectly wetted impact surfaces. The microscopic fly ash from pulverized bituminous coal and from anthracite coal defeats nearly all air cleaning devices except the most effective types of viscous coated metal impact surfaces. The lint and fibrous material frequently encountered in the air in clothing and textile manufacturing districts tends to

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clog up the air passages of most types of viscous coated filters and makes it advisable to use highly specialized air cleaning devices in such districts. The unoiled, paper-like or fabric-like types of air cleaning devices serve best in these circumstances.

The manufacturer of air cleaning devices therefore must use his shot gun efficiency guarantee with mental reservations, and, lest he get himself into trouble, has been for some time endeavoring to devise, with the help of research, some uniform and assuredly accurate method of testing and rating.

There is a fairly satisfactory method now in use for testing and rating fans. Within the author's memory, however, there was a time when most fan manufacturers rated their product on hope and guess rather than on knowledge and truth. If the ordinary contractor who must purchase a fan knew his business, he knew that the published fan rating was like a list price, subject to a discount of anywhere from 30 to 60 per cent, and even then it would not be unusual for the contractor to be compelled to insert intermediate blades in the fan wheel or to increase the fan speed and the motive power somehow, before he could escape the responsibilities of his guaranteed air delivery.

This situation concerning fans happily has been cleared up following the adoption of a Standard Test Code for Disc and Propeller Fans, Centrifugal Fans and Blowers, and the ultimate consumer, the contractor, the engineer and the manufacturer all have benefited thereby.

The same situation concerning the rating of air cleaning devices shrieks for resolution and to be fair in the matter it must be stated that the manufacturers of air cleaners are cooperating fully in this movement.

How to measure the dust in air, so as to evaluate air cleaning devices for general ventilation work, is the great question. Sticky surfaces such as pitch-coated, ruled glass slides can have some air impacted against them out of a syringe; then the sticky surfaces can be placed under a powerful microscope. The tiny area of the microscope's cognizance will show visible dust spots or dust colonies, and these can be counted. If one little ruled area of the sticky glass has 100 dust particles, then the whole exposed sticky area probably has 100 particles for each similar ruled subdivision, perhaps 10,000 particles per square inch. If one fraction of a cubic foot of the air forced against the sticky surface discloses 10,000 dust particles, probably each cubic foot has the same dust burden and if there are 1,000,000 cu ft per hour passing by, perhaps each one of these million cubic feet will have the same dust burden. The counting of the dust particles through a powerful microscope, it has been observed, cannot easily be duplicated by different people. One man will see 110 particles, while another will see 100, and another will see only 85, even though all use the same apparatus and the same ruled area. An error of seemingly trifling dimensions in the microscope count will be magnified to almost unbelievable and to thoroughly unacceptable ranges so that this method of testing air cleaning devices for rating purposes is therefore open to question.

A second method of determining the amount of dust in air, employs a tube of fine material such as sugar. A sample of dusty air is drawn slowly through the sugar, which strains out the dust. The sugar can be weighed before and after its dust-catching experiences and the weight of dust caught from a given volume of the air is an index of the probable amount of dust in each cubic

foot of the main air stream. This method of dust determination has an advantage over the visual counting method in that any number of men can weigh the same article on an accurate scale and all will agree. The scale is more accurate than the eye-brain combination. The optimist has less chance to play

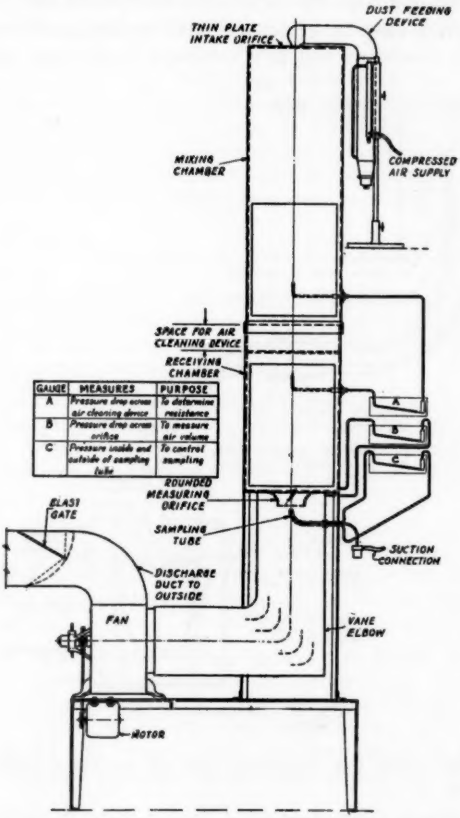

Fig. 1. Testing Apparatus for Air Cleaning Devices

on his hopes with a scale-balance weighing to a ten thousandth of a gram than with the inconsistent human eye. The sugar of course may pick up moisture from the air, or may lose it to the air while catching the dust.

A third method draws a sample of the dusty air through a filter paper. The paper clogs and interposes increasing resistance as the dust accumulates. This increase in resistance, gauged with air volume, evaluates the dustiness.

A fourth type of dust determinator brings about contact by a measured sample of the dusty air with a surface which is sticky, and gauges the dust by weighing the tube or the plate which supports the sticky substance.

Another dust-collecting scheme uses an electric precipitator for a measured fraction of the main stream, and weighs the precipitated dust.

There are a great many other types of dust determinators. Many of them under laboratory conditions will give consistent results and will repeat their

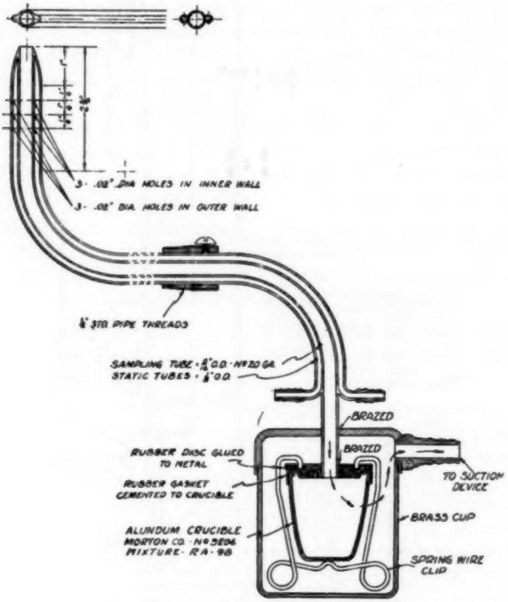

FIG. 2. SAMPLING TUBE

performances day after day provided that all of the conditions day after day are the same.

The various Society committees which have labored to produce a Code for Testing Air Cleaning Devices, however, have concluded that before a code could be effective an approved dust determinator must be selected. Their labor of many years has not been rewarded with a shot-proof apparatus.

Is it not now apparent that dust in air is of such extreme variability that a limited determinator for rating air filters is impracticable?

Will it not promote the best interests of all concerned if this fact is recognized and agreement is reached upon the point of testing for rating of air cleaning devices for general ventilation work by means of some non-proprietary scheme, of reasonably consistent performance, within the dust

ranges commonly encountered in ventilating systems? If an affirmative answer to these two questions can be obtained it will simplify very greatly the labors of the Committee on Atmospheric Dust and Air Cleaning Devices.

In recent months it has been the author's privilege to make hundreds of tests of air cleaning devices under laboratory conditions. The tests were made for comparative purposes and covered about 100 different types of alleged air cleaners.

The work was started, using the proposed code of the committee, which was presented for study at the June 1932 meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS at Milwaukee, Wis. As the work progressed various changes in the apparatus and methods proved desirable. Since perhaps not more than 20 members of the Society are familiar with the subject, the details which are outlined here may be interesting.

DESCRIPTION OF TEST APPARATUS

The testing apparatus, Fig. 1, shows the arrangement for using the weight method of determination. The air tunnel is vertical, to reduce the tendency of the heavier dust particles to fall out of the air stream before they reach the air cleaning device or the sampling tube.

With a horizontal air tunnel such precipitation of the heavier particles is evident in the mixing chamber. The agglomerated dust masses which break away from the device being tested, between it and the sampling tube, also are much in evidence in any horizontal receiving chamber to leeward of the air cleaning device.

In the dust sampling tube, Fig. 2, the orifice faces the cleaned air stream. When the pressure is balanced across the Gauge *B* the air velocity induced by the pump suction connection through the crucible and the Orifice *O* must be the same as that maintained by the exhaust fan in the duct into which the Orifice *O* is inserted.

Fig. 3 is one form of dust feeding device. The bottom of the dust hopper has a long fine screw thread so that the device easily may be rotated one revolution or so, back and forth. The clearance between the end of the compressed air Pipe *A* and the cup is adjustable by the Screw *B*. The stirring rod, attached to the cup, rotates with it and thus breaks up any arching over of the dust. Manipulation of the dust hopper bottom only occasionally is necessary.

The question, "How to measure the volume of air which passes the air cleaning device?" is of prime importance.

It is not too easy to measure the volume of air passing through the air-cleaning device from which the sample which will be tested for dust shall be taken, nor is it easy to measure accurately the volume of the sample of dust-carrying air itself. The oldest scheme for air velocity measurement and the only one available now for certain limited purposes, involves using an anemometer. An anemometer of course is only a geared wind wheel which expresses the air-flow speed. If the instrument is in good condition; if it is held squarely to the plane of the air current; if the air current is not too fast

and if it is not too slow to come within the scope of the particular anemometer; if the transverse cross section area of the air-way is measured accurately at the plane in which the anemometer is placed; if the stop watch is accurate; if the reading of the dials and of the stop watch are coordinated perfectly; then the anemometer method will approximate the volume of air in the main air-way.

In general, however, engineers do not accept anemometer readings as more than approximate, though they are of great value for relative determinators when adjusting and equalizing the air flow through the different ventilating outlets and inlets in any given building.

If, in a straight air-way or liquid-way, there shall be interposed a slight reduction in area followed by a return to normal area, and if the air or liquid

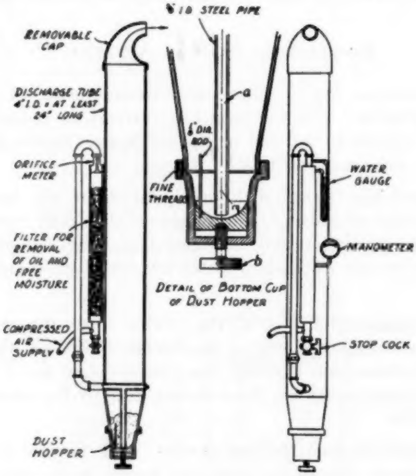

FIG. 3. DUST FEEDER

is being forced at a reasonable speed along the channel, the volume of the air or liquid which is passing can be computed with great accuracy provided that the following factors are known: the area of the passageway at the point of maximum restriction, and the difference between the static pressure up-stream and that down-stream from the reduction, or orifice. This orifice, or venturi tube method of measuring gases or liquids, is readily installed and is exceedingly reliable and accurate, and as its use reduces liability of human error to a great extent, the method is standard for laboratory determinations. It is also the preferred method for a great many field observations.

Air volume flowing along a duct also may be measured by a pitot tube. This method is preferable to the anemometer method but requires much higher velocities than those which often occur in ventilating systems. The tube compares the pressure of the air moving along the duct against atmospheric pressure

outside of the duct and if these pressures are taken by traversing the tube in many short stages across the air-way, the computed velocities may be averaged to an accurate result. The venturi tube method, however, in a test set-up for rating air cleaning devices is very much to be preferred to the pitot tube method since the former set-up may be permanent and the influence of the human element is minimized. The operator merely reads the scale, just as he does with the dust weighing arrangement.

How shall the volume of the air samples taken from the main air-way be measured, before making dust content analysis?

Small volumes of air under appreciable pressure can be measured accurately and practically by a wet gas meter. This method has been employed successfully by the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in measuring infiltration through window cracks and in measuring the air volume expelled by steam from the interiors of pipes and heat-transfer surfaces. If the air sample to be analyzed for dust content, however, is a fair one, taken from the main air stream and if the air flow speed through the sampling tube is maintained accurately at the same rate as in the main air stream, there will be no need of any quantitative measurement of the air sample by means of a gas meter. For instance, if the main stream air velocity is 1000 fpm and if the cross sectional area is 4 sq ft, 4000 cfm are passing by.

If the sampling tube is ¼ in. diameter (0.0003 sq ft) but has the same 1000 fpm speed as the main air stream, it will be passing 0.3 cu ft every minute. All that is necessary for assurance in this connection will be that the dust separating device will pass this much air at the pressure available. The sampling tube is provided with two auxiliary tubes which parallel the main sampling tube near its intake. One of these opens into the interior of the sampling tube while the other opens into the main airway.

The outer ends of these two auxiliary tubes are connected each with the other through a water gauge, and the suction pressure through the sampling tube is to be regulated so that this balancing gauge stands at zero.

How shall the difficulties encountered with sugar and similar dust-filtering media, that may be unstable due to changing atmospheric conditions, be overcome?

There are available in the chemical laboratory supply market standard burned clay filter-crucibles. These are made of well mixed ingredients in uniform size, thickness and porosity. They are porous to the flow of gases. They can be provided with soft rubber gaskets around their rims and when these crucibles are placed over a sampling tube they do permit air flow through their walls while retaining the dust out of the air. The more dust they hold, of course, the more pressure will be required to force air through them, but if several pounds pressure as by a positive blower or by a piston type air pump are available, all necessary pressure can be maintained throughout many hours of heavy testing with one crucible.

In the practice followed in these laboratory tests for rating of an air cleaning device, a uniform volume of air is moved past the sampling tube for a timed period without any air cleaning unit in place. The crucible during this period receives its quota of the air and of the air borne dust. If dust, accumulating

against the windward side of the crucible, tends to obstruct the air flow through the sampling tube the pressure gauge between the two auxiliary or balancing tubes will indicate lack of balance and the observer must correct this by increasing the suction on the leeward side of the crucible.

When testing an air-cleaning device for rating, it is not practicable to wait around perhaps for weeks while the resistance builds up with the normal air dust. It is necessary, and assuredly permissible, to accelerate the process by introducing dust artificially. This dust need not be natural dust, because natural dust is not uniform but varies at different levels and at different city neighborhoods and in different cities. It is desirable therefore to use some standard dust which will be more or less representative of normal air impurities, which can be compounded anywhere and which shall serve as a part of the universal measuring stick of air cleaner efficiency, for general ventilation work.

After much discussion and experimentation, the dust of simple character which most nearly simulates actual city air and at the same time is universally obtainable appears to be one composed of 50 per cent by weight of common powdered lampblack, and 50 per cent by weight of bituminous coal ashes, which will pass a 200 mesh sieve. Any air cleaning device which will stop this aggregate will stop any kind of tangible iniquity which can be borne by an air stream.

The Bureau of Standards has developed a method of testing air filters which uses as the medium the output from an electrostatic precipitator for flue dust.

The dust must of course not just be dumped into the air but must be fed in a uniform manner during the entire period of test and must simulate the procedure and concentrations of nature, so that if tests are being made with a wetted air-cleaning device, for instance, the water will be given enough time to saturate each succeeding particle of dust. This time element is equally important for a viscous coated air-cleaning device.

The dust, in a receptacle, tends to arch itself over any releasing orifice and when the arch eventually breaks there is an undesirable dust cloud. There are recorded in the TRANSACTIONS OF THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, several practical types of automatic dust mixing and feeding devices.

Since compressed air must be used in this proposed test why not employ it to deliver the dust?

An automatic dust feeding device was developed; one of several effective types, in which a stream of compressed air under constant pressure of several pounds through a very small orifice may dislodge the dust particles and may inject them into the main air stream, much as a similar stream of compressed air would be used to inject pulverized coal into the feeder of a combustion device. After a little practice the rate of flow of the compressed air can be regulated so as to consume any desired length of time in delivering any desired quantity of dust.

An air cleaning device may remove dust from air, but may not do it at an acceptable resistance to the passage of the main air stream. It is apparent that if any air cleaning device does arrest dust and if it holds the dust which it arrests, so delivering a continuously improved air stream as to dust content,

the airways through the cleaning device tend eventually to clog up. From the moment when any non-automatic air cleaning device begins to receive dusty air it must begin to interpose increasing resistance to air flow with every minute that passes, until its time for reconditioning shall arrive.

There is no better way of indicating the resistance to air flow through a dust filter than by windward and leeward static pressure tubes, connected to a gauge which registers the pressure difference, just as did the tubes and gauge described for the sampling tube. If the ducts to windward and to leeward of the air cleaning device are not identical in size, then total pressure tubes, opening toward the direction of air flow, should be substituted for static tubes.

As a summary of the foregoing dissertation, with the methods and the instruments outlined, a reasonably careful and patient person may determine by cut and try methods for any air cleaning device, the best combination of the following factors:

1. Volume of air per unit of presentation area to which the apparatus is best adapted. This will be governed by the resistance to air flow and by the ability of the device to arrest and to retain dust at that resistance-point.
2. Resistance to air flow will be governed by the amount of air forced through the device and by its ability at this speed to maintain a high arrestance of dust. The resistance to air flow of a non-automatic air cleaning device and probably the resistance of all air cleaning devices changes constantly from the moment of the beginning of dust arrestance to the moment when reconditioning commences.

   Such devices should be rated at the conditions of volume, resistance and arrestance in effect when the resistance is half way between its dustless minimum and its dustiest maximum.
3. Arrestance of dust will be established by the average conditions in effect when a uniform volume of air is being handled over the entire useful life of the device, this useful life being limited by the resistance to air flow.

Thus an air-cleaning device might be rated at: 300 cfm per sq ft of face area at:

Resistance; clean 0.10 in.; resistance at end of reconditioning cycle 0.46 in.; average resistance 0.22 in.

Arrestance, average during test period of 10 hours, 94 per cent. This is obtained by the following formula: Arrestance $= 1 - \frac{G_1}{G_0}$, in which G_1 is the dust caught by the sampling instrument to leeward of the air cleaning device and G_0 is the dust caught under the identical conditions with the air cleaning device removed.

No longer should an air cleaning device be selected merely because an efficiency of 97 per cent or 98 per cent is claimed. The method should be to select one proved to have a capacity something like the following: To arrest 94 per cent of standard dust at an average resistance to air flow not exceeding

0.22 in., from 300 cfm of air per square foot of face area. (Cell face, not necessarily filter face area.)

If the device is of the cell type select one which is guaranteed to hold this dust at the specified efficiency (meaning by efficiency in this case both arrestance and resistance) for the longest time. If the device is of the automatic type the rating would be in the same wording, with perhaps a greater volume of air per minute per square foot of face area.

The apparatus described and illustrated herein is substantially that of the Proposed Code for Testing and Rating Air Filters presented at the Semi-Annual Meeting, 1932, at Milwaukee. Modifications in detail of crucible mounting and preference for a vertical testing duct are the only changes in the physical set-up.

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No. 961

A PIPE SIZER FOR DETERMINING THE SIZES OF PIPES AND OF RESTRICTING ORIFICES IN A HOT-WATER HEATING SYSTEM

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RECENTLY there has been a noticeable increase in the use of hot water heating systems, especially in large buildings. The advantages of this kind of a system, where room temperatures are easily regulated by changing the temperature of the water, have always made it a popular type, particularly in residences. The simplicity is such that expert care is not required.

Not until the last few years, however, has there been sufficient information available regarding the flow of hot water in pipes to enable the designer to proceed intelligently, and he has been forced to use rules-of-thumb rather than rational formulas. The research work done by Prof. F. E. Giesecke and associates at the Texas Engineering Experiment Station, of the Agricultural and Mechanical College of Texas, in cooperation with the A. S. H. V. E. Research Laboratory, has provided this information, so that now such systems may be laid out scientifically.

The purpose of the design of the Pipe Sizer, which will be described, was to take this available information, including that regarding friction through orifices, and place it in such form as to make it readily usable with the least amount of labor.

It is not necessary at the present time to discuss the general principle of operation of a hot water system, since there is no change proposed here in the method of calculation; but only a simplification of the means by which the available pressure heads may be balanced by friction heads so as to produce even circulation throughout the system.

Sources of Data

In the A. S. H. V. E. Guide of 1925-26 there are two logarithmic charts (See Figs. 33 and 34, pp. 80 and 81) showing Friction Heads in Milinches per Foot of Pipe and per Elbow respectively. In each case the heat conveyed

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per hour is in thousands of Btu, with a difference in temperature of the water in the flow and return lines at 20 F.

In the A. S. H. V. E. GUIDE, 1933 (Table 5, p. 98) is given the Friction Heads (in Milinches) of Central Circular Diaphragm Orifices in Unions for various sizes of orifices in pipes from ¾ in. to 2 in. inclusive. In this table the velocity is given in inches per second.

The logarithmic charts from which this table was prepared, together with a table showing orifice friction in 2½ in. pipe, have been furnished by W. H. Badgett, of the Texas Engineering Experiment Station.

FEATURES DESIRED

In order to produce a device having as wide a scope as possible it was felt that the following points must be considered:

1. Some arrangement so that temperature differences other than 20 F could be used. (The Pipe Sizer has been arranged so that this difference, or temperature drop, may be taken at any value from 5 F to 30 F inclusive.)
2. Make the design such that the load carried by a pipe may be considered either in thousands of Btu per hour or equivalent square feet; so that the user may employ whichever unit is most convenient for him.
3. Instead of reading *unit* frictions in pipe and fittings, provide a means so that the *total* friction may be read without the necessity of multiplying the unit friction by the number of units.
4. Incorporate the information regarding friction heads in orifices in such a way that it may be easily applied, and without the necessity of calculating the velocity.

DESCRIPTION OF DEVICE

As will be noted in Fig. 1, there are five parts to the device which was designed: (1) Body, (2) Disk, (3) Pipe Sector, (4) Fitting Sector, and (5) Orifice Sector.

1. *Body:* At the bottom there are two scales from 5 to 30, both showing Temperature Drop Through Radiator (F); at the left a scale from 1 to 100 showing Length of Pipe (feet); at the right a scale from 1 to 100, showing Number of Equivalent Elbows; and at the top an arrow marked Pipe Size. In the upper right corner is also a convenient Table of Equivalent Elbows.
2. *Disk:* Along the lower edge is a scale from 1 to 10,000, showing Load Carried By Pipe, expressed either in Thousands of Btu or Equivalent Sq Ft, whichever the user finds more convenient. If the load is to be considered in thousands of Btu, the scale opposite the bottom left scale on the body is used; and if in equivalent square feet, that opposite the bottom right scale is used.

   On the left side of the disk are Curves For Pipe sizes from ½ in. to 6 in.; on the right side are corresponding Curves For Fittings. The use of these is described in the following paragraphs 3 and 4.

   Along the top edge of the disk is a scale from 50 to 100,000, showing

Friction Head In Orifice in milinches and its use is described in paragraph 5.

3. *Pipe Sector:* This sector, on the left side of Pipe Sizer, and projecting over the disk, carries an arrow marked Length of Pipe, which is to be set opposite Length of Pipe In Feet on body. Along its upper edge is a scale showing Friction Head in Pipe in milinches; readings are taken where Curves For Pipe on disk intersect this edge.
4. *Fitting Sector:* This sector, on the right side of the Pipe Sizer, is similar to that on the left, and similarly used. The readings show Friction Head in Fittings in milinches.
5. *Orifice Sector:* This sector, which extends under the disk, carries on its right edge a scale of pipe sizes from ¾ in. to 2½ in., which is to be set opposite the arrow of Pipe Size on body. A set of curves showing Orifice Diameter (inches), from 0.3 to 1.6, are on the face of this sector. With the pipe size used set opposite the arrow of Pipe Size on body, opposite the Friction Head in Orifice to be produced (shown at top of disk), read the Orifice Diameter to be used in that size of pipe.

Example

If 50 sq ft of hot water radiation is carried by a 1 in. pipe 8 ft long with 10 equivalent elbows and the temperature drop of the water in passing through the radiator is 20 F, what is the friction head produced? If the pressure head available is 510 mil in., what size orifice must be inserted in the 1 in. pipe to use up the remaining head?

Solution (See Fig. 1). Set 50 sq ft (on the lower edge of disk) opposite 20 F drop (on bottom right scale on body); set arrow on pipe sector at left opposite 8 (length of pipe in feet); set arrow on fitting sector at right opposite 10 (equivalent elbows); then along upper edge of pipe sector read where curve for 1 in. pipe intersects, 55 mil in. and along upper edge of fitting sector read where curve for 1 in. fitting intersects, 140 mil in.; total 195 mil in. friction.

With a pressure head of 510 mil in. and a friction head of 195 mil in. there is left 315 mil in. to be produced by an orifice. Leaving disk in same position (with 50 sq ft opposite 20 F drop), on orifice sector set 1 in. (diameter of pipe) opposite pipe size arrow on body; then opposite 315 mil in. friction (on upper edge of disk) read 0.54 in. (diameter of orifice).

In this example the load was expressed in sq ft. If preferred, thousands of Btu may be used. The equivalent of 50 sq ft (at 150 Btu per sq ft per hr) is 7.5 thousands Btu. Then set 7.5 (on the lower edge of disk) opposite 20 F drop (on bottom left scale on body), and proceed as before. *Note that this is the same setting of disk as before.*

It is of course possible to provide the necessary friction by a choice of pipe sizes, and not use any orifice. If a certain size of pipe produces too much friction, and the next size too little, a portion of the pipe may be chosen of one size, and the remainder of the other, and thereby make the total friction equal the value desired.

The same example will illustrate the effect of different pipe sizes. To de-

termine how much of each size would be necessary to produce a friction head to balance the 510 mil in. of pressure head, note the following tabulation:

Length of Pipe	No. of Equiv. Ells	Size of Pipe	Friction Head		
			Pipe	Fittings	Total
8	10	1 in.	55	140	195 (1 in., too large)
8	10	¾ in.	200	400	600 (¾ in., too small)

By trying a portion of this as 1 in., and the remainder as ¾ in., it is found that:

Length of Pipe	No. of Equiv. Ells	Size of Pipe	Friction Head		
			Pipe	Fittings	Total
2	2	1 in.	14	25	39
6	8	¾ in.	150	320	470
8	10	....	....	....	509 (satisfactory)

Where the friction head to be produced is about the average of the friction of a pipe which is too large and the next size which is too small, one size may be used as the supply, and the other as the return. In general, however, it will be found easier to figure the orifice sizes.

With a given load on a certain size pipe, only one setting of the disk is required to determine friction of pipe, fittings, and orifice.

It is not necessary at any time to calculate the velocity of water.

METHOD PURSUED IN DESIGN

Since the water carried by each section of pipe varies directly as the heating load and inversely as the temperature drop through the radiator, these two factors may be laid out logarithmically, the temperature drop through the radiator being placed on the Body of the Pipe Sizer at the bottom and the load along the lower edge of the Disk. The graduations for temperature drop are laid off twice: *first,* opposite the load in thousands of Btu; *second,* opposite the load in square feet, assuming the customary heat emission of 150 Btu per sq ft per hour.

The charts shown in THE GUIDE, 1925-26 (pp. 80 and 81) are similar, except that on page 81 the slope of all the lines is the same and has a value of 2, showing that the friction in fittings varies as the square of the velocity. These lines when placed on the Disk become a series of curves of exactly the same shape, the only change being that from rectangular to polar coordinates. On page 80 the lines for pipe sizes have a gradually increasing slope from the larger to the smaller sizes. When these lines are placed on the Disk, using polar coordinates, they become a series of similar curves, but somewhat fan-shaped.

The Pipe Sector and Fitting Sector, each overlapping the disk, are provided with logarithmic graduations; so that when the arrow on the sector is placed opposite the number of units, the intersection of the curves of pipe or fitting

diameters with the edge of the sector give the total friction for that number of units directly.

In devising a method of showing the friction heads in orifices, it was necessary to determine not only how that friction varied with the load, but also, if possible, to see what relation existed between the friction in various orifice diameters in each of the various sizes of pipe. From an inspection of the charts for orifice frictions it could readily be seen that the friction varied in each case as the square of the velocity, all of the lines on all charts being

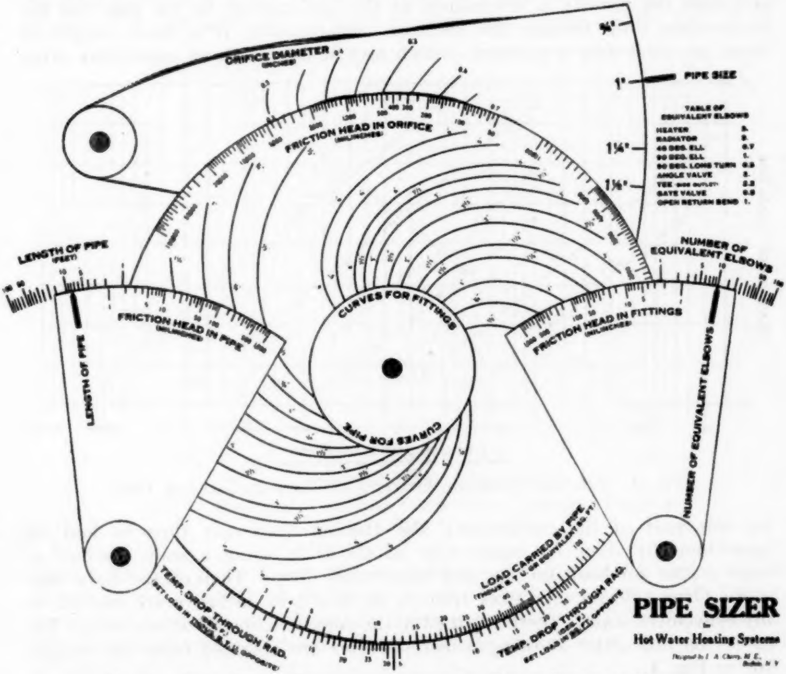


Fig. 1. Pipe Sizer, Showing Setting to Solve Problem Given

parallel and having a slope of 2. It was not immediately apparent, however, just what the relationship was between the friction in various sized orifices in any size of pipe. The figures taken from the charts were plotted in a number of ways, the most illuminating of which is that shown in Fig. 2, which is semi-logarithmic, the abscissae being logarithmic and the ordinates arithmetic. This shows the loss of head in mil inches, when handling a load of 1000 lb of water per hour, passing through orifices whose diameters are of varying ratios to the diameter of the pipe. The investigations covered a range of ratios of diameter of orifice to diameter of pipe from ⅓ to ⅔, values beyond these figures not being consistent.

The general appearance of the lines on the chart in Fig. 2 suggested the

making of Fig. 3. This chart is also semi-logarithmic, but instead of considering the same load in pounds of water per hour for all sizes of pipe, a constant velocity of 4 in. per second was taken. In this case all of the points fell very close to each other along a single straight line. This simplified the design of the computer, as it showed that for a certain ratio of diameter of orifice to diameter of pipe, and with a certain velocity, the friction was practically the same, irrespective of the pipe size.

Since the friction heads in the orifices vary as the square of the velocity, and since the velocity is determined by the load carried by the pipe and the temperature drop through the radiator; consequently, if a fixed weight of water per cubic foot is assumed (which may be done without appreciable error

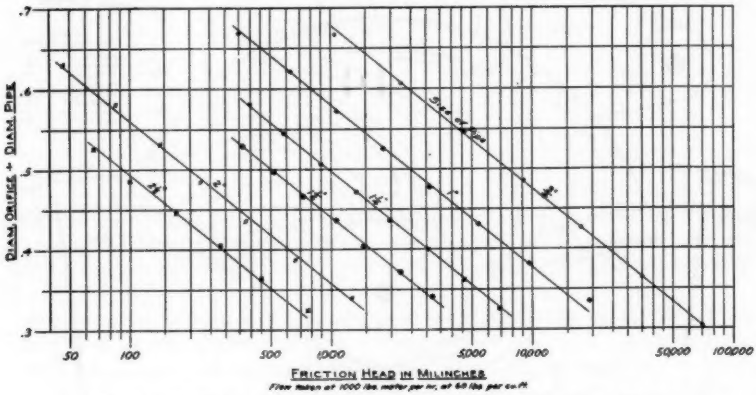


FIG. 2. FRICTION THROUGH ORIFICES IN VARIOUS SIZES OF PIPE

for this part of the calculation), the friction head may then be laid off logarithmically along the upper edge of the Disk, using a scale one-half as large as that for loads on pipe and temperature drop. Then on the right edge of the Orifice Sector pipe sizes from ¾ in. to 2½ in. inclusive are laid off, to any convenient scale. These are used in the determination of orifice sizes. The curves on this sector showing orifice diameter are obtained from the straight line in Fig. 3.

CALCULATION OF TWO-PIPE UPFEED DIRECT RETURN GRAVITY SYSTEM

The calculation sheet (Fig. 4) shows a complete piping plan, with boiler, pipe, and radiators, and it is proposed that the pipe sizes be chosen so that the system will operate with a temperature drop of 25 F (190 F to 165 F). The calculations for the system are given completely on the sheet, in a form arranged for this purpose. (Obviously the item letters and guide lines are not used in actual practice, but only to describe the method of procedure.)

From the Chart of Pressure Heads (Fig. 5) it will be noticed that the dotted line shows that with an initial temperature of 190 F and a final temperature of 165 F, a head of 108 mil in. per ft of height will be produced. From the building or plans it is found that the height from the middle of

the boiler (between the grate and top) to the middle of the first floor radiators is 5 ft; to second floor radiators, 15 ft; and to third floor radiators, 24 ft. The resulting pressure heads are as indicated in the upper corner of the sheet, Fig. 4.

Calculation of Mains

The main to be considered first is the one supplying the least favored radiator, which in this case is the 80 sq ft radiator at the end of the longer main. This radiator is the least favored, because it has the smallest pressure head (being on the first floor) and the greatest friction head (being at the end of the

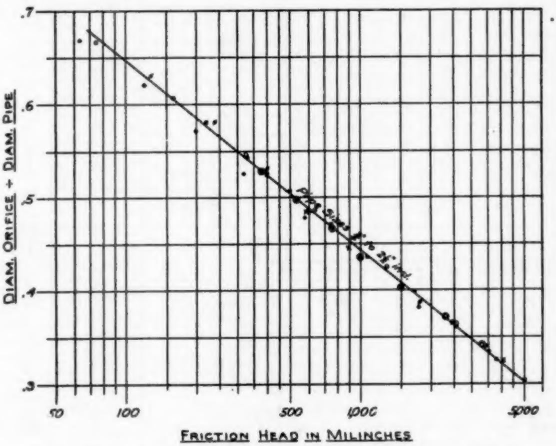
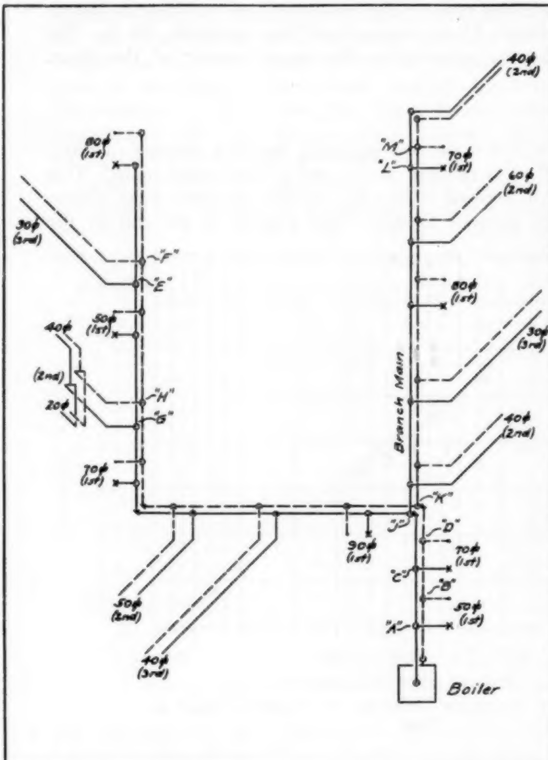


FIG. 3. FRICTION THROUGH ORIFICES IN VARIOUS SIZES OF PIPE

line). Starting at the boiler and in order along this main list under mains, the quantities of radiation expressed in square feet (or in thousands Btu, if preferred) as shown in Column *A*. For demonstration sq ft has been chosen. The third figure in this column (Item *a*) is the branch main, carrying a total of 320 sq ft.

Starting at the bottom of Column *B* and working upward, following the guide line, the continued totals are recorded which give the load carried by each section of the main. The total of all radiation is 910 sq ft (the top item in Column *B*).

In Column *C* enter the length of pipe in each section of the mains (including both supply and return as one item) opposite the load for that section. From the plan, including vertical pipes also (which are not shown) it is found that the length of the supply main from the boiler to *A* is 10 ft, and of the return to *B* is 16 ft; adding these the sum 26 is the length of pipe carrying 910 sq ft. If in the supply main, the distance from *A* to *C* is 5 ft, and in the return from *B* to *D* is the same, enter 10 as the length carrying 860 sq ft, and so on for the other lines. For the pipe carrying 80 sq ft (the last radiator)



Gravity Hot Water Heating System

INDUSTRIAL PLANNING CORP.
CONSULTING ENGINEERS
BUFFALO, N.Y.

Date
Job No
Archt.
Owner
Buildi.

MAINS						
Loads sq. ft.	Total Loads sq. ft	Pipe Length ft.	Equiv. Ells no.	Pipe Size in.	Friction Pipe mil.in.	Friction Fittings mil.in.
A	B	C	D	E	F	G
50	910	26	9	3½	70	150
70	860	10	–	3½	23	–
Branch 320a	790	8	2	3½	17	24
90	470	10	–	3	16	–
40	380	16	–	2½	50	–
50	340	8	–	2½	21	–
70	290	14	2	2½	27	15
60	220	10	–	2½ / 2	18 / 22	–
50	160	12	–	2	19	–
30	110	6	–	1½	15	–
80	80	30	12	1½	40	44
	Totals	150	25			
		125 / 275	5 / 125			
Branch Main						
40	320a	8	6	2½	19	55
30	280	12	–	2½	21	–
80	250	14	–	2½	22	–
60	170	10	–	2	17	–
40	110	6	–	1½	15	–
70	70	12	14	1½	26	80
	Totals	62	20			
		80 / 142	4 / 80			

FIG. 4. CALCULATION SHEET FOR A TWO-PIPE

include the length of the radiator runouts and the stubs up to the middle of the radiator, as these are parts of this circuit.

In Column D enter the total equivalent ells in each section of main, as given in table on the Pipe Sizer. In the first section of this system there are 6 ells and the boiler (3 equivalent ells); so that 9 is the number of equivalent ells in this section. Similarly enter the equivalent ells for the remaining sections. In the last section, carrying 80 sq ft, include the equivalent of the radiator valve (2 ells) and of the radiator itself (3 ells), which with the ells in the piping makes a total of 12.

The pipe sizes of the main will now be selected approximately, and then corrected where necessary. The total length of this circuit is 150 ft (Column C), with 25 equivalent ells (Column D). For a rough estimate, assume that an elbow is equivalent to twice as many feet of pipe as the diameter of the pipe in inches: *i.e.*, a 1 in. ell equivalent to 2 ft of 1 in. pipe; a 2 in. ell equivalent to 4 ft of 2 in. pipe; and so on. Assuming that the average size

July 30, 1932			PRESSURE HEADS (IN MILINCHES)		
118	**WATER**	Location	Height above Boiler in ft.	Head per ft.	Total Pressure Head
John Jones	Entering Rad. 190 deg.	1st Fl.	5	× 108 =	540
Buffalo, N.Y.	Leaving Rad. 165 deg.	2nd Fl.	15	× 108 =	1620
William Smith.	Temp. Drop. 25 deg.	3rd Fl.	24	× 108 =	2592
Buffalo, N.Y.					
ig Store and flat.					

Heads			BRANCHES AND RISERS													
								Pressure Heads			Friction Heads					
Section Totals mil. in.	Cont. Totals mil. in.		Loads sq. ft.	Pipe Length ft.	Equiv. Ells no.	Pipe Size in.	Floor	Total mil. in.	Used in Mains mil. in.	Available for Branch mil. in.	Pipe mil. in.	Fittings mil. in.	Pipe and Fittings mil. in.	Remainder for Orifice mil. in.	Diameter of Orifice in.	
H	J		K	L	M	N	P	Q	R	S	T	U	V	W	Y	
220	220		50	10	14	1 ~~3/4~~	1	540	220	320	45 ~~750~~	125 ~~350~~	170 ~~500~~	150	0.57	
23	243		70	10	14	1	1	540	243	297	85	225	310	−13	—	
41	284_b															
16	300		90	12	14	1¼	1	540	300	240	40	130	170	70	0.85	
50	350		40	46	18	¾	3	2592	350	2242	470	280	750	1492	0.32	
21	371		50	30	18	¾	2	1620	371	1249	450	450	900	349	0.46	
42	413		70	16	14	1¼	1	540	413	127	35	80	115	12	—	
~~12~~ 27	425		60_c	30	12	1			425		200	150	350			
			20_d	12	10	¾	2	1620	775_k	845	32	40	72	773	0.26	
			40_e	16	12	¾	2	1620	775_m	845	170	190	360	485	0.4	
19	444		50	16	14	1½	1	540	444	96	19	39	58	38	0.77	
15	459		30	46	18	¾	3	2592	459_f	2133	260	170	430	1703	0.26	
84			80	30	12	1½	1	540	459_g	81	46	44	84	−3	—	
543 ~~368~~																
284_b	284_b															
74	358		40	30	18	¾	2	1620	358	1262	300	280	580	682	0.37	
21	379		30	46	18	¾	3	2592	379	2213	260	170	430	1783	0.26	
22	401		80	12	14	1½	1	540	401	139	35	100	135	4	—	
17	418		60	30	18	1	2	1620	418	1202	200	220	420	782	0.45	
15	433		40	50	16	¾	2	1620	433_h	1187	500	250	750	437	0.41	
106			70	12	14	1¼	1	540	433_j	107	26	80	106	1	—	
539																

UPFEED GRAVITY HOT WATER HEATING SYSTEM

of this main is 2½ in., the equivalent length of pipe per ell is 5 ft, or 125 ft for the 25 ells, making the total equivalent length 275 ft, as shown. The pressure head to produce the flow to the first floor is 540 mil in., so the total friction head must not exceed this figure, or $540 \div 275 = 2$ mil in. per ft of length (approximately).

Now use the Pipe Sizer for calculations. Set 910 sq ft (the total load) on disk opposite 25 F drop on body; set arrow on pipe sector opposite 1 on body, since the drop is in mil inches per foot of length; then opposite 2 mil inches on edge of sector, note that the nearest curve is that for 3½ in. pipe. In Column *E* enter 3½ in. as a trial size for this pipe. Continue down this column, successively setting the various total loads opposite 25 F drop, and choosing the size of pipe whose curve is nearest to the friction of 2 mil inches.

With these trial sizes (Column *E*), and with lengths of pipe and number of fittings in each section (Columns *C* and *D*), the actual friction head will now be determined, and then such changes made in pipe sizes as are necessary to

make the friction head equal to the pressure head (See Fig. 6). Set 910 sq ft on disk opposite 25 F drop on body; set arrow on pipe sector opposite 26 (length of pipe) on body; set arrow on fitting sector opposite 9 (number of equivalent ells); then where curve for 3½ in. pipe on disk intersects edge of pipe sector, read 70 mil in. as the friction in that section of pipe (Column *F*); and where curve of 3½ in. fitting on disk intersects edge of fitting sector, read 150 mil in. as the friction due to the fittings in that section (Column *G*); and enter their total of 220 for that section (Column *H*). Continue for each load and size of pipe, using the respective lengths and number of fittings. Adding Column *H*, the total friction head is 558 mil in.; but since only 540 mil in. of pressure head is available, the friction must be reduced so as not to exceed this figure. This is effected by increasing the size of one or more sections of pipe. It makes no difference just where this is done, but since the reduction to be made is small, it can be easily done in a short length of pipe. Try increasing the pipe carrying 220 sq ft, from 2 in. to 2½ in.; then the friction of this section of pipe is reduced from 27 mil in. to 12 mil in., and the total from 558 mil in. to 543 mil in., which is satisfactory. (There are no fittings in this section.)

Now fill in Column *J* of the continued totals, by adding successively the section totals from Column *H*, as shown by guide lines. These give the total of the frictions in both supply and return mains from the boiler up to the branch connection to each radiator, and which will be used later.

In calculating the branch main (Item *a*) and its branches, the condition is somewhat different, as the least favored radiator on this line is not the last one, but is the last of those on the first floor, the 70 sq ft. The circuit to this least favored radiator includes the mains from *M* to the boiler and back to *L*, and the radiator branches from *L* to the 70 sq ft radiator and back to *M*.

If the assumption had been made that the last radiator on the second floor was the least favored, the second floor head would have been considered as available for circulation. The result would have been smaller mains and a friction head in the complete circuit to the 70 sq ft radiator on the first floor, greater than the pressure head available for circulation. This 70 ft radiator would not function properly. Therefore it is necessary to use sizes which will be satisfactory for the first floor radiator, the least favored on this line.

In the same manner as before, fill in Columns *A*, *B*, *C* and *D*. The last two radiators are apparently reversed, but as the circuit to the 70 sq ft radiator is being considered as the least favored, the 40 sq ft radiator is reached first.

The total length of this circuit is 62 ft; assuming the average size of pipe to be 2 in. and each ell equivalent to about 4 ft of pipe, or 80 ft for the 20 ells, the total is 142 ft. The total pressure head for this first floor radiator is 540 mil in., of which 284 mil in. (Item *b*) is used in the main from *K* to the boiler and back to *J*, so that 256 mil in. is available for use in the portion from *J* to *L* to the 70 sq ft radiator to *M* to *K*. Since the equivalent length of this part is 142 ft, the drop is $256 \div 142 = 1.8$ mil in. per foot of length (approximately).

Set 320 sq ft opposite 25 F drop on body; set arrow on pipe sector opposite 1 ft of length; opposite 1.8 mil in. on top of sector, note that the nearest curve is 2½ in. (Column *E*), the size of main for this section. Similarly obtain sizes of the remaining sections.

With these trial sizes, and the length of sections of mains and the number of equivalent ells, calculate the frictions as outlined before. The total friction in this circuit is 539 mil in., which is almost identical with the pressure head

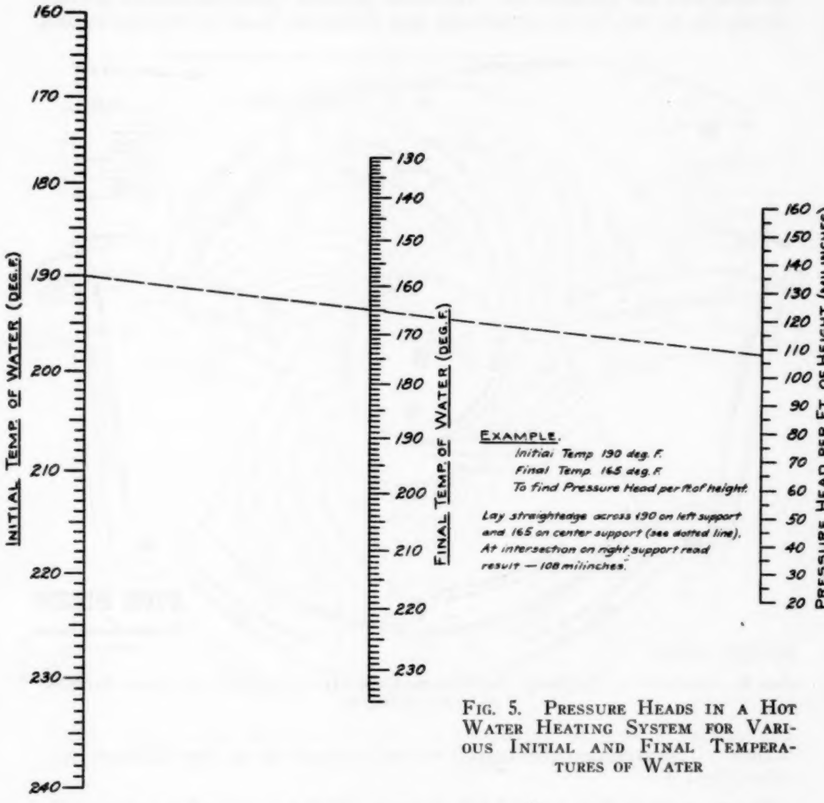


FIG. 5. PRESSURE HEADS IN A HOT WATER HEATING SYSTEM FOR VARIOUS INITIAL AND FINAL TEMPERATURES OF WATER

of the 540 mil in., and is satisfactory. Now fill in the column of continued totals (Column *J*).

Calculations for Branches and Risers

Enter the individual loads in Column *K* as in Column *A*, except that the branch main (Item *a*) is omitted; Item *c* being the riser, and Items *d* and *e* the two radiators on this riser. Fill in Columns *L* and *M* with the lengths of pipe and equivalent ells, as given on plans, using values of equivalents from table on the Pipe Sizer. Fill in Columns *P* and *Q*, showing the floor on which each radiator is located, and the corresponding pressure head, taken from upper part of calculation sheet. Column *R* is the same as Column *J*, *except*

where more than one radiator is on a branch; also note that Items *f* and *g* (Column *R*) are alike, similarly Items *h* and *j* (Column *R*).

In the first line subtract 220 (Column *R*) from 540 (Column *Q*), and enter the difference 320 (Column *S*). This is the pressure head remaining to produce circulation to the 50 sq ft radiator, and should be used in friction in this

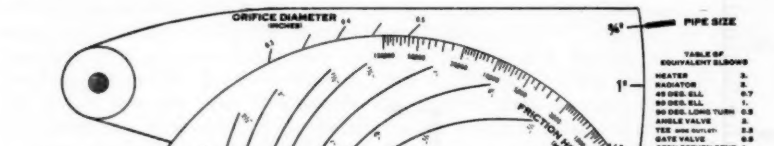


Fig. 6. Pipe Sizer, Showing Setting to Determine Friction in First Section of Main (Fig. 4)

branch. Try assuming the branch to this radiator at ¾ in. (Column *N*). (See Fig. 7.)

Set the load of 50 sq ft on disk opposite 25 F drop, arrow on pipe sector opposite 10 (Column *L*), and arrow on fitting sector opposite 14 (Column *M*), then read at the intersection of ¾ in. pipe curve the value of 150 (Column *T*), and at the intersection of ¾ in. fitting curve 350 (Column *U*), making the total 500 (Column *V*). The friction head of 500 is greater than the available pressure head of 320 (Column *S*), and is not satisfactory. Without changing setting of Pipe Sizer, read for 1 in. a pipe friction of 45 mil in., and a fitting friction of 125, total 170, which is less than the 320 mil in. available. Without moving disk on Pipe Sizer, set 1 in. on orifice sector opposite arrow, and opposite friction head of 150 (Column *W*) on top of disk read diameter of orifice of 0.57 in. (Column *Y*). Similarly calculate the items for the remaining radiators, except the two on one riser, using pipe size just large enough so

that the friction of pipe and fittings will be less than the pressure head, and supplying remaining friction by an orifice.

In figuring the sizes of pipes for the two radiators on one riser, the simplest method is partly by trial. Assume riser to be 1 in. (Column N) and calculate 200 (Column T), 150 (Column U), and 350 (Column V), this being the total

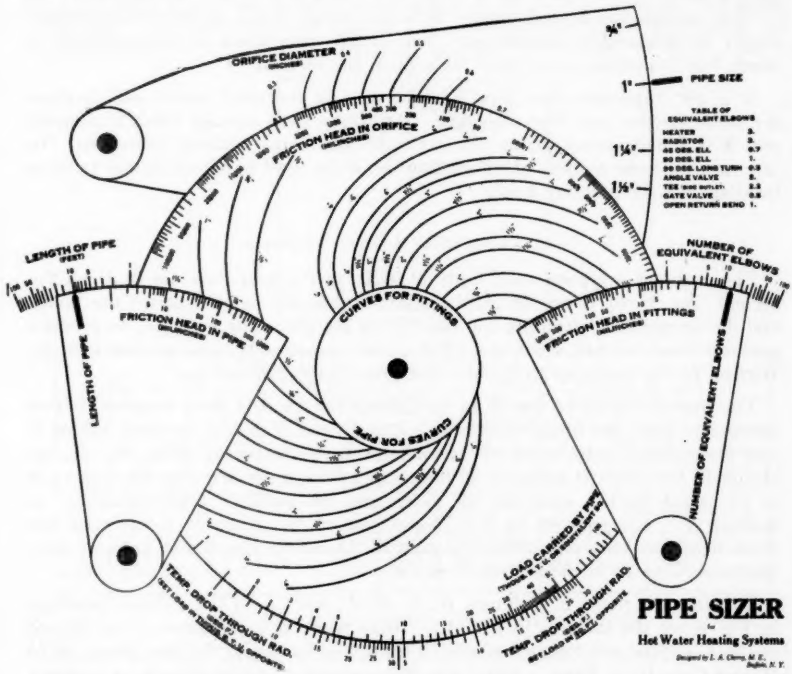


Fig. 7. Pipe Sizer, Showing Setting to Determine Size of Branches and Orifice for 50 Sq Ft Radiator Nearest Boiler (Fig. 4)

drop in the risers alone. Add this figure to the 425 mil in. (Columns J and R) which is the drop in the mains from H to the boiler and back to G, and the total is 775 mil in., which is now put down as Items k and m in Column R. Subtract 775 (Column R) from 1620 (Column Q) and enter the difference 845 (Column S), which is available for the friction in each of these two radiators and its branches. Try the pipe size for the 20 sq ft radiator at ¾ in. and calculate friction and orifice.

This pipe size might have been taken at ½ in., in which case the pipe friction would have been 130 mil in.; fitting friction 130 mil in.; and the total, 260 mil in., which is well below the available pressure head of 845 mil in. (Column S). But as there are no data available at present on orifices in ½ in. pipe, ¾ in. is used. Calculate the size of pipe for the 40 sq ft radiator on this riser

in a similar manner, as shown. The remaining items on the calculation sheet require no explanation.

Fig. 4 then gives all of the information necessary for the completion of the plans, and shows a system completely *balanced*. All pipe sizes have been calculated, and the sizes of all orifices, where necessary, are given.

This example has been described in rather minute detail, so that the procedure might be thoroughly understood. The actual calculation is accomplished in much less time than is necessary to explain the method.

It is not imperative that pipe sizes be used in the main which will produce the same friction per foot of length. Arbitrary sizes may be used, if desired, and it is found advantageous oftentimes to use mains without reductions. In any case the same method of calculation should be used in balancing the friction head against the pressure head.

CALCULATION OF A TYPICAL RISER

Fig. 8 shows a typical riser with 50 sq ft on the first floor, 40 sq ft on the second, and 35 sq ft on the third. Assume that the heights above the boiler and the temperature drop are the same as in the preceding example, to produce pressure heads of 540, 1620, and 2592 mil in. respectively; also assume that the friction in the mains up to the tees for this riser is 230 mil in.

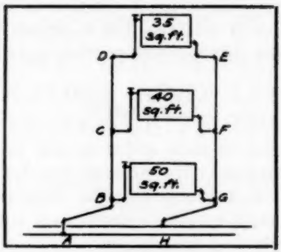
The branch circuit to the 50 sq ft radiator on the first floor consists of two sections of pipe; the branch and riser (*A* to *B*, and *G* to *H*), carrying 125 sq ft and the radiator connection (*B* to radiator to *G*) carrying 50 sq ft. In the circuit to the 40 sq ft radiator on the second floor there are also the risers (*B* to *C*, and *F* to *G*), carrying 75 sq ft, and the radiator connections (*C* to radiator *F*), carrying 40 sq ft. The circuit to the 35 sq ft radiator on the third floor has only one additional section, including risers and radiator connections (*C* to *D* to radiator to *E* to *F*).

Tabulate the items in Columns *K*, *L*, *M*, *P*, and *Q*. (The column headings in Fig. 8 are the same as in Fig. 4.) Items *a* and *b* in Columns *L* and *M* are the feet of pipe and number of ells in branch and riser to the first floor (*A* to *B*, and *G* to *H*). Items *c* and *d* are for connections to the first floor radiator (*B* to radiator to *G*). Items *e* and *f* are for risers from first to second floor (*B* to *C*, and *F* to *G*), and Items *g* and *h* are for connections to second floor radiator (*C* to radiator to *F*.) Items *j* and *k* include risers from second to third floor, and radiator connections (*C* to *D* to radiator to *E* to *F*). Item *m* in Column *R* is 230 mil in. according to the conditions stated in the problem and Item *n* in Column *S* is 310, the difference between 540 (Column *Q*) and 230 (Column *R*).

The size of branches to risers, and connections to the first floor radiator, must have a friction head that will approximate this available pressure head of 310 mil in. Try the branch size at 1½ in. and calculate pipe friction 50 (Column *T*), fitting friction 75 (Column *U*), or a total 125 (Column *V*). The friction head up to the tees (*i.e.*, *B* to boiler to *G*) for the first floor radiator is then found by adding 230 (Column *R*) and 125 (Column *V*) to make 355 (Item *p*), leaving 185 (Item *q*) to be used in friction in the connections to this radiator (*B* to radiator to *G*). Assume radiator connections to be 1 in.

and calculate frictions and orifice size for the 50 sq ft first floor radiator in the usual manner. The orifice diameter is 0.66 in.

The friction head to tees B and G is 355 (Item r) and is the same as Item p. As the total pressure head for the second floor is 1620 mil in. the available pressure head is 1265 (Item s), the difference between the two. Then the friction from B to C to radiator to F to G must be such as to equal this available head. Try this section of riser at 1 in., and calculate frictions, totaling 100 (Column V). Then Item t is found by adding 355 and 100, or 455. Item u



BRANCHES AND RISERS													
Loads sq. ft.	Pipe Length ft.	Equiv. Ells no.	Pipe Size in.	Floor	Pressure Heads: Total mil. in.	Pressure Heads: Used in Mains mil. in.	Pressure Heads: Available for Branch mil. in.	Friction Heads: Pipe mil. in.	Friction Heads: Fittings mil. in.	Friction Heads: Pipe and Fittings mil. in.	Friction Heads: Remainder for Orifice mil. in.	Diameter of Orifice in.	
K	L	M	N	P	Q	R	S	T	U	V	W	Y	
125	16_a	8_b	1½		540	230_m	310_n	50	75	125			
50	6_c	12_d	1	1	540	355_p	185_q	28	100	128	57	0.66	
75	10_e	$-_f$	1		1620	355_r	1265_s	100	—	100			
40	6_g	12_h	¾	2	1620	455_t	1165_u	60	190	250	915	0.35	
35	29_j	10_k	¾	3	2592	455_v	2137_w	230	130	360	1777	0.29	

FIG. 8. CALCULATION SHEET FOR A TYPICAL RISER

therefore is 1165. Taking this section of risers at ¾ in., calculate frictions and orifice size for the 40 sq ft second floor radiator. The orifice diameter is found to be 0.35 in.

Similarly Item v is 455, and Item w is 2137. A trial using ¾ in. for the pipe to this third floor radiator and calculating remaining items, gives the orifice for this radiator 0.29 in.

CHART OF PRESSURE HEADS

In order to facilitate the calculation of pressure heads, Fig. 5, already mentioned, was plotted. The formula for the pressure head is as follows:

$$h = \frac{a - b}{\frac{a + b}{2}} \times 12 \times 1000$$

where

h = Head (mil in.)
a = Weight of water at final temperature (pounds per cubic foot)
b = Weight of water at initial temperature (pounds per cubic foot)

This equation may be transformed so as to read

$$\frac{a}{b} = \frac{24{,}000 + h}{24{,}000 - h}$$

The design of this alinement chart presented no unusual difficulties, but a mathematical curiosity appeared. Since this chart is to perform a division, a and b are laid off logarithmically; however, the resultant pressure head appears with graduations which are apparently arithmetic, though actually logarithmic. In the equation, h is always very small relative to 24,000, so that within the limits in question the value of this fraction is proportional to the logarithm of the value, consequently the graduations on this part of the chart are equally spaced.

CONCLUSION

The writer does not claim to have added in any way to the existing amount of information on this subject. The attempt has been made to take research data and arrange them in such a way that the labor involved in the calculation of a hot water heating system may be reduced to a minimum, and so encourage the use of a scientific method of calculation.

It will be apparent that the calculation of other types of systems may be facilitated, including one-pipe gravity, forced circulation, or combinations of the two.

It might be added that this device has been used not only to design a number of gravity hot water heating systems, but also to determine all changes to be made on some existing *trouble jobs* so that they functioned properly.

DISCUSSION

L. W. MOON: If the pipe sizer is used in determining friction or resistance based on the length of run and the number of fittings as shown on the plans, and then in the actual installation it is found necessary to use a greater number of fittings than contemplated by the designer, due to structural conditions, would the increased resistance tend to upset the calculations and necessitate changes in the orifices?

In the case of an old job, where it is desired to correct existing faults and where risers and runouts are concealed in walls and floors, would it be possible to determine accurate orifice sizes notwithstanding the inability to determine the amount of pipe and exact number of fittings thus concealed?

L. A. CHERRY: By using a greater number of fittings in an installation than contemplated, the resistance is increased, and if a sufficient number of them are introduced the system will not be well balanced and will not work as well. However, a few additional fittings will not make any material difference.

In an old job, where risers and runouts may be concealed, the problem is more difficult. It then becomes necessary, by noting the sizes of the pipes, where they disappear into the walls or floors, and their location, to determine as nearly as possible the lengths and sizes of the pipes in these unknown portions, and to estimate the number of fittings in such lines. While it is impossible to get the exact lengths and the exact number of fittings, these quantities may be obtained with sufficient accuracy to enable the resistance to be figured. After the calculations are completed, the necessary changes may be made in the pipe sizes, or orifices may be introduced to balance the system. This has proved satisfactory in a number of cases.

No. 962

THE APPLICATION OF THE EUPATHEOSCOPE FOR MEASURING THE PERFORMANCE OF DIRECT RADIATORS AND CONVECTORS IN TERMS OF EQUIVALENT TEMPERATURE

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This paper is the result of research sponsored by the American Society of Heating and Ventilating Engineers and conducted at the University of Illinois

The data presented in this paper were obtained in connection with an investigation conducted by the Engineering Experiment Station of the University of Illinois, of which Dean M. S. Ketchum is the Director. This work is carried on in the Department of Mechanical Engineering. The results will ultimately comprise part of a bulletin of the Engineering Experiment Station. Acknowledgment is due E. L. Broderick, Research Assistant in Mechanical Engineering, for valuable assistance in the collection and preparation of the data.

IN a previous paper [1] it was shown that a temperature of 68 F maintained at the 30-in. level in a room is a more acceptable standard for specifying the room temperature at which direct radiators and convectors shall be tested under service conditions, than 70 F maintained at the 60-in. level. It was further shown that when 68 F was maintained at the 30-in. level, any differences in heating effect, or effectiveness of the radiators or convectors in providing for human comfort, were directly reflected in differences in the steam condensation obtained. In this case, the temperatures were measured by means of thermocouples of No. 22 B. & S. gage wire, and the question arose as to whether any modification in the conclusions would become necessary

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§ Special Research Assistant Professor in Mechanical Engineering, University of Illinois.
[1] Steam Condensation an Inverse Index of Heating Effect, by A. P. Kratz and M. K. Fahnestock, A. S. H. V. E. Transactions, Vol. 37, 1931, p. 475. Also Reprint No. 1, Engineering Experiment Station, University of Illinois.

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Hotel Statler, Detroit, Mich., June, 1933, by A. P. Kratz.

if a sizable instrument more nearly approximating the proportions of the human body were used as an index of the combined effects of radiation and convection, instead of the comparatively small thermocouples that were actually used. Such an instrument, designated as an eupatheoscope,[2, 3, 4] has been developed by the Building Research Board in England. A similar instrument was, therefore, constructed, and further tests were run to determine whether the steam condensation of direct radiators and convectors could be correlated with the *equivalent* temperature as indicated by, and explained in connection with, the eupatheoscope, and whether any correlation existed between the read-

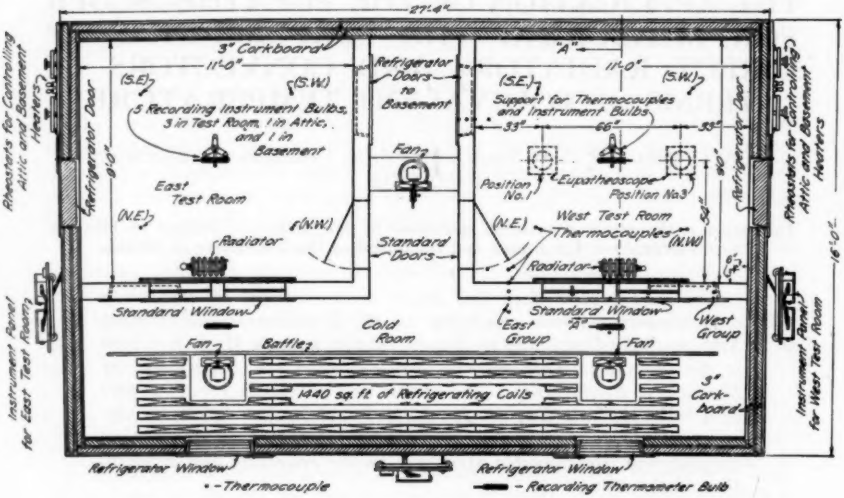

FIG. 1. PLAN SECTION OF LOW-TEMPERATURE TESTING PLANT

ings of the eupatheoscope and the dry-bulb temperature indicated by means of thermocouples made from No. 22 B. & S. gage wire.

DESCRIPTION OF APPARATUS

All tests were conducted in the West test room of the room heating testing plant. The latter has been completely described in previous publications.[1, 5, 6, 7]

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[2] Department of Scientific and Industrial Research, *Report of the Building Research Board for the Year 1929*, His Majesty's Stationery Office, London, England.

[3] The Eupatheostat, by A. F. Dufton, *Journal Sci. Inst.* 1929, 6, (8), pp. 249-51.

[4] The Equivalent Temperature of a Room and Its Measurement, by A. F. Dufton, Department of Scientific and Industrial Research, *Building Research Technical Paper* No. 13, 1932, His Majesty's Stationery Office, London, England.

[5] University of Illinois Engineering Experiment Station *Bulletins Nos.* 192 and 223.

[6] Investigation of Heating Rooms with Direct Steam Radiators Equipped with Enclosures and Shields, by A. C. Willard, A. P. Kratz, S. Konzo, and M. K. Fahnestock, A. S. H. V. E. TRANSACTIONS, Vol. 35, 1929, pp. 79-89.

[7] Performance of Convector Heaters, by A. P. Kratz and M. K. Fahnestock, A. S. H. V. E. TRANSACTIONS, Vol. 38, 1932, p. 351.

The location of the various instruments, and of the radiators or convectors as tested, is shown in the sections in Figs. 1 and 2. The thermocouples, borne by the standard at the center of the room, were made from No. 22 B. & S. gage wire and were placed at the levels shown in Fig. 2. A study of the temperature conditions within the test room proved that the center reading at a given level was representative of the average temperature at that level.

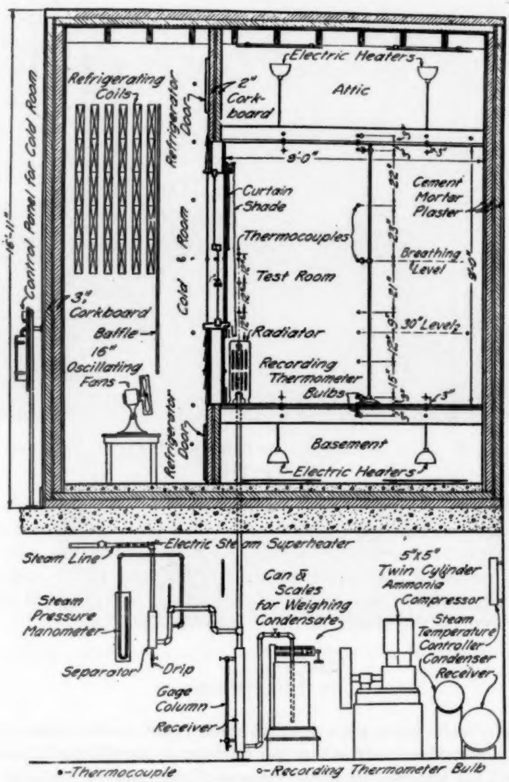

FIG. 2. ELEVATION SECTION OF LOW-TEMPERATURE TESTING PLANT

Most of the tests were run with the exposed walls containing a door and a window as shown in Figs. 1 and 2. For a few tests, however, storm sash and a storm door were installed, and sheets of ½-in. insulating wall board were attached to the window and door frames on the inside of the room, thus

eliminating the window and door, and giving the effect of two unbroken exposed walls.

The direct cast-iron radiators used in these tests consisted of (1) a 9-section, 25-in., three-tube radiator having 1½ in. between the centers of the sections, (2) a 12-section, 26-in., three-tube radiator, having 2½ in. between centers, (3) an 8-section, 26-in., five-tube radiator, having 2½ in. between centers,

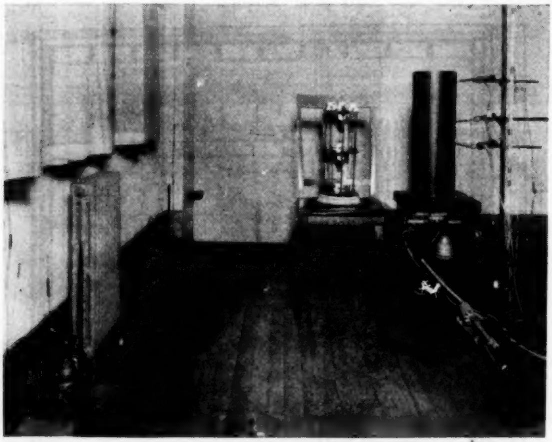

FIG. 3. THE EUPATHEOSCOPE WITH HEATING ELEMENT REMOVED

and (4) a 3-section wall type radiator, each section having a nominal rating of 7 sq ft. The convectors consisted of (1) three convectors with non-ferrous heating units, including two representative types, and (2) a convector with a cast-iron heating unit. The principal dimensions of the radiators and convectors are given on the insets in Figs. 6 to 9.

THE EUPATHEOSCOPE

The principle underlying the eupatheoscope involves the measurement of the heat loss from a sizable body when the surface is maintained at some constant temperature. This implies the measurement of the amount of heat, or electrical current, required to maintain the surface at this same constant temperature.

The main element of the eupatheoscope used on these tests consisted of a hollow copper cylinder 7½ in. in diameter and 22 in. long, placed vertically on a stool so that its mean height was 30 in. above the floor, as shown in Fig. 3. Twelve thermocouples, made from No. 34 B. & S. gage copper and constantan wire, were embedded flush with the surface of the cylinder proper, which was blackened, and one was embedded flush with the surface of the top. These 13 thermocouples were electrically but not thermally insulated from

the cylinder and were connected in series to a recording-controlling potentiometer, shown in Fig. 4, thus affording means of both indicating and controlling the average temperature of the surface of the cylinder.

In order to heat the surface, current from the line was supplied to a lamp-bank placed inside of the cylinder. This lamp-bank, removed from the cylinder, is shown in Fig. 3. The lamps, arranged as shown, in order to give uniform

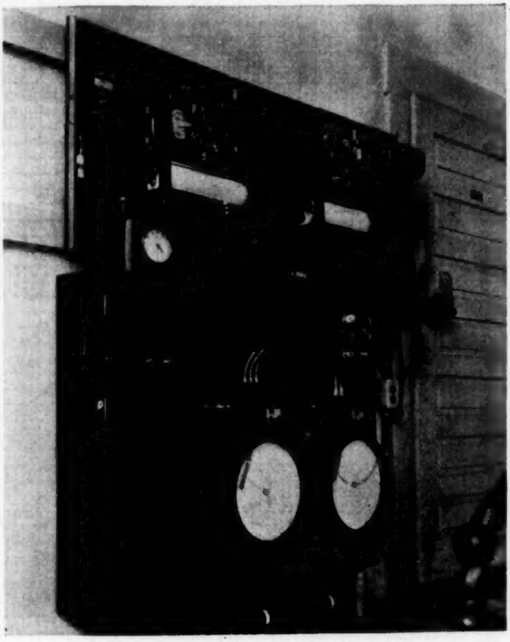

FIG. 4. RECORDERS AND CONTROLLER FOR THE EUPATHEOSCOPE

surface temperature, were connected in two separate circuits. Part of the lamps were supplied with a constant amount of current just insufficient to maintain the surface at the required temperature. Current to the rest of the lamps was supplied intermittently through the action of the recorder-controller, thus maintaining the surface at the required constant temperature. The total current supplied was measured by means of a Lincoln thermal converter (Fig. 4) connected to a recording potentiometer. This arrangement proved satisfactory, but the necessity for using a step-up current transformer introduced difficulties in calibration that warrant the recommendation of a low range recording wattmeter for use in connection with similar eupatheoscopes.

Since human comfort is directly correlated with the rate of heat loss from the

body, the eupatheoscope affords a means of evaluating probable comfort in terms of the heat required to maintain the cylinder at some constant temperature. The criterion provisionally adopted by the Building Research Board [2, 4] in England was based on observations indicating that in a comfortably warmed room a sedentary man loses by radiation and convection, neglecting evaporation as subsequently explained, approximately 17.5 Btu per square foot per hour,

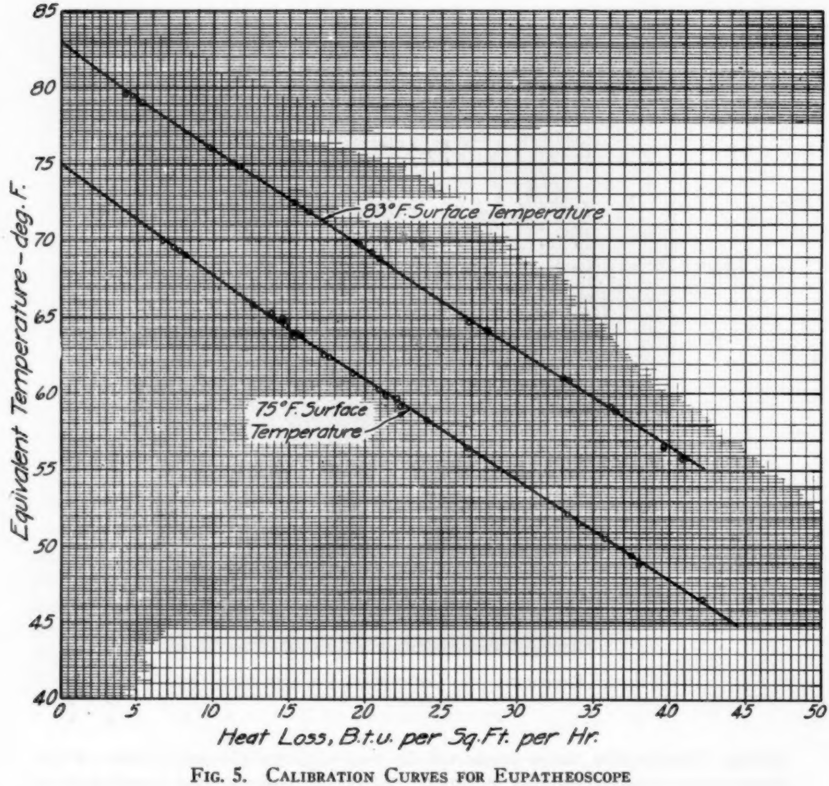

FIG. 5. CALIBRATION CURVES FOR EUPATHEOSCOPE

and that the average temperature of surface of the body and clothing is approximately 75 F. A room was, therefore, assumed to be comfortably warmed when the eupatheoscope loses heat at the rate of 17.5 Btu per square foot per hour with the surface temperature maintained at 75 F. This heat loss in a non-uniform environment may result from various combinations of radiation and convection, and, from the standpoint of comfort, it has been assumed that it is the rate of heat loss that is important, and not the relation existing between radiation and convection. Hence, the equivalent temperature of a given non-

uniform environment has been defined as the temperature of a uniform enclosure, with walls and air at the same temperature, in which, in still air, a sizable black body at 75 F would lose heat at the same rate as it does in the non-uniform environment.

In its practical application, therefore, the eupatheoscope was calibrated by subjecting it to various temperatures in a uniform environment, in still air, with walls and air at the same temperature, and determining the rate of heat loss with the surface of the cylinder maintained at 75 F. These equivalent temperatures plotted against the rates of heat loss are shown in the lower calibration curve in Fig. 5. The heat losses observed in non-uniform environment may then be referred to this curve in order to express the combined effect of radiation and convection in the given environments in terms of the equivalent temperatures that would be required in a uniform environment to produce the same heat losses as those observed. The lower calibration curve shown in Fig. 5 agreed within approximately 1.5 F with the one published for the instrument used by the Building Research Board.[2]

It may be observed that if a heat loss by radiation and convection of 17.5 Btu per square foot per hour with a surface temperature of 75 F is accepted as the criterion of comfort, the equivalent temperature conducive to comfort is 62.5 F. This, of course, corresponds to the dry-bulb temperature and not to the effective[10] temperature. Even at high relative humidities, a dry-bulb temperature of 62.5 F cannot be regarded as representative of American ideas or standards of comfort. Hence, it seems advisable to adopt a different definition of equivalent temperature to serve as a provisional standard more nearly conforming to American practice.

From the work of F. C. Houghten[8] and others, it has been established that the heat loss by radiation and convection from an adult, seated at rest in still air with 20 per cent relative humidity, is approximately 300 Btu per hour. For an average man, with 19.5 sq ft of body surface, this is equal to 15.4 Btu per square foot per hour. Furthermore, from the work of L. B. Aldrich,[9] in which the measurements were made both with a modified form of bolometer and with thermocouples, the average mean temperature of the surface of the skin and clothing on a number of adults was determined as approximately 83 F. Hence, the eupatheoscope, as used in this investigation, was recalibrated with the surface temperature maintained at 83 F. The results of this calibration are shown in the upper curve in Fig. 5. From this curve it may be observed that with a heat loss by radiation and convection of 15.4 Btu per square foot per hour, the corresponding equivalent temperature is 72.3 F.

Extensive investigations made by the A. S. H. V. E. Research Laboratory and summarized in a report by C. P. Yaglou,[10] have established the fact that for the average individual, at rest in still air under normal conditions of temperature and relative humidity, an effective temperature of 66 F is most

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[8] Thermal Exchanges Between the Bodies of Men Working and the Atmospheric Environment, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant, *The American Journal of Hygiene*, Vol. XIII, No. 2, March, 1931, p. 425.

[9] A Study of Body Radiation, by L. B. Aldrich, *Smithsonian Miscellaneous Collections*, Vol. 81, No. 6, Dec. 1, 1928.

[10] How to Use the Effective Temperature Index and Comfort Chart. Report of the Technical Advisory Committee on Re-Study of Comfort Chart and Comfort Line, by C. P. Yaglou, W. H. Carrier, E. V. Hill, F. C. Houghten, and J. H. Walker, A. S. H. V. E. TRANSACTIONS, Vol. 38, 1932, p. 410.

conducive of comfort. The term effective temperature is not synonymous with equivalent temperature. However, since the measurements were made in rooms in which the temperature of the walls was practically the same as that of the air, the dry-bulb temperatures observed are identical with the equivalent temperatures as defined for the use of the eupatheoscope. With a relative humidity of 20 per cent, an effective temperature of 66 deg corresponds to a dry-bulb temperature of 73 F, and with a relative humidity of 40 per cent to a dry bulb temperature of 71 F. Hence, over the range of relative humidities practical for the average residence, an effective temperature of 66 deg agrees reasonably well with the equivalent temperature of 72.3 F shown by the upper calibration

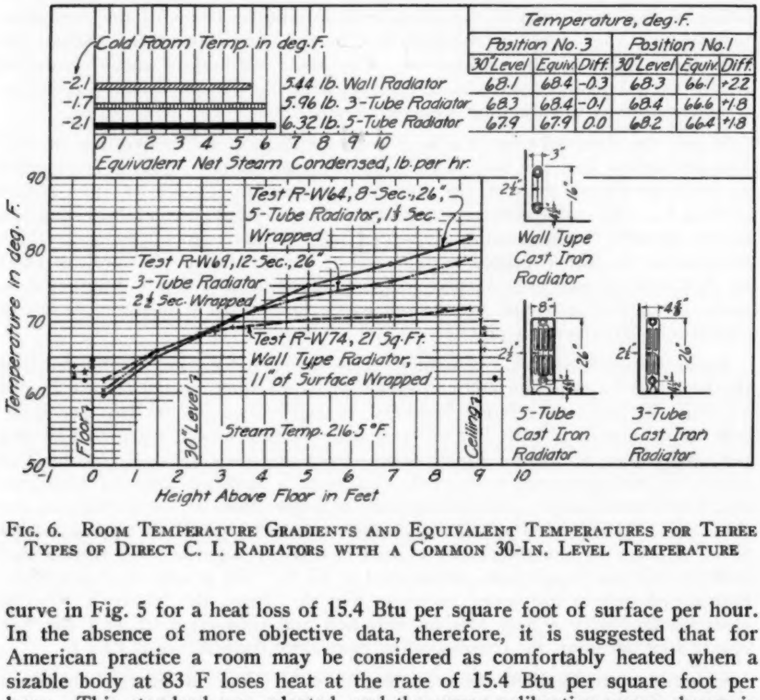

FIG. 6. ROOM TEMPERATURE GRADIENTS AND EQUIVALENT TEMPERATURES FOR THREE TYPES OF DIRECT C. I. RADIATORS WITH A COMMON 30-IN. LEVEL TEMPERATURE

curve in Fig. 5 for a heat loss of 15.4 Btu per square foot of surface per hour. In the absence of more objective data, therefore, it is suggested that for American practice a room may be considered as comfortably heated when a sizable body at 83 F loses heat at the rate of 15.4 Btu per square foot per hour. This standard was adopted, and the upper calibration curve shown in Fig. 5 was used for all of the work on radiators and convectors reported in this paper. Since the two calibration curves are related through the equivalent temperature, the results can be converted to the English standard, if such a conversion is desired.

Since the eupatheoscope primarily determines the heat loss by radiation and convection alone, it is evident that some account must be taken of the effect of relative humidity in interpreting the readings. However, at normal relative humidities and at temperatures less than 75 F, small changes in relative

humidity may be neglected in comparison with other factors. The work of F. C. Houghten [8] and others indicates a decrease of only 25 Btu in the total heat loss from an adult when the relative humidity changes from 20 per cent to 95 per cent at a constant dry-bulb temperature of 72 F. The relative humidity in the test room during the tests on radiators and convectors remained at approximately 15 per cent. Hence while interpretations of the results in terms of a standard of absolute comfort might be subject to modification, the readings of the eupatheoscope may be accepted as an accurate index

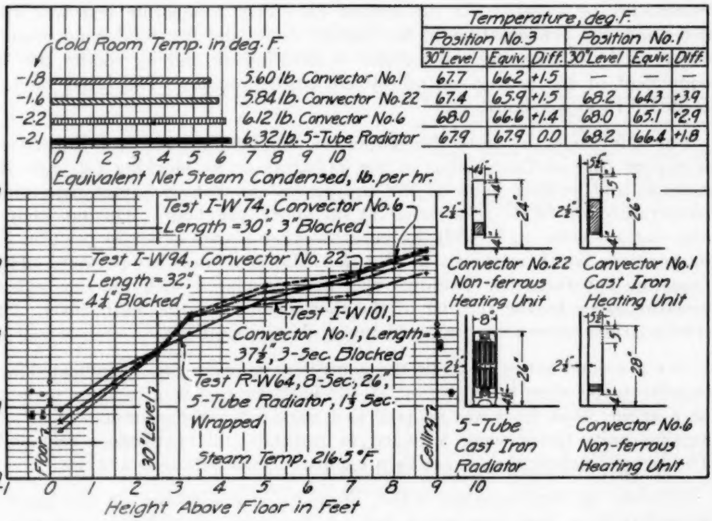

FIG. 7. ROOM TEMPERATURE GRADIENTS AND EQUIVALENT TEMPERATURES FOR A DIRECT C. I. RADIATOR AND THREE CONVECTORS WITH A COMMON 30-IN. LEVEL TEMPERATURE

of the relative performance of the different radiators and convectors judged on the basis of the environment established. The standard suggested is only provisional and somewhat arbitrary. A fruitful field of research is open in establishing an absolute standard in the light of actual physiological requirements, in determining whether the relation between radiation and convection is important in regard to human comfort, and in determining the modifications made necessary by changes in relative humidity. These considerations were definitely beyond the scope of the investigation of the performance of radiators and convectors.

TEST PROCEDURE

In all cases, the temperature of the cold room was maintained at about −2.0 F, and one of the exposed walls was subjected to an equivalent wind velocity of approximately 10 miles per hour. The temperature of the air

above the ceiling of the test room was maintained at 62 F and the air in the basement at such a temperature that the upper surface of the floor was approximately two degrees warmer than the lower surface. No test observations were made until conditions had remained constant for several hours, as indicated by the readings of the thermocouples on the inside wall surfaces. This required about 20 hours after a change in radiators or convectors was made. When the required thermal constancy had been attained, the condensate was weighed over the period of one hour, and no test was accepted if the condensate showed more than 2.5 per cent deviation in the successive 10-min increments of weight. At the end of each test, a separate test was run to determine the condensation in the piping alone, and the total condensate was corrected by subtracting the amount so determined, and by adding the steam equivalent of the heat given off by the eupatheoscope. The latter never exceeded 0.11 lb per hour.

For one series of tests the output of the heating unit was adjusted, by wrapping part of the sections of the radiators or by blocking off part of the cabinets and heating units of the convectors, so that each unit maintained a temperature of 68 F at a level 30 in. above the floor. During each test, observations were made with the eupatheoscope placed either in position No. 1 or position No. 3 as shown in Fig. 1. The equivalent temperature obtained from the eupatheoscope readings was then compared with the dry-bulb temperature given by the reading of the thermocouple at the 30-in. level. Steam condensations were also compared under these conditions.

For a second series of tests, the equivalent temperature as indicated by the eupatheoscope when the 8-section, 26-in., five-tube cast-iron radiator maintained 68 F at the 30-in. level was adopted as a standard, and the outputs of all of the heating units tested were adjusted to maintain this equivalent temperature. The steam condensations were then compared under these conditions.

In order to determine the effect of doors and windows on the equivalent temperature, a few tests were run on one type of radiator and one type of convector, for which the door and window were blocked as previously indicated.

RESULTS OF TESTS

The results of all of the tests are shown in Figs. 6 to 9. The tabulated values for the temperature at the 30-in. level with the eupatheoscope in position No. 3 all correspond with values read from the curves, which were established simultaneously with the eupatheoscope readings. Those for position No. 1 correspond with temperature gradients established on different tests at the same time that eupatheoscope readings in position No. 1 were made. Since it was impossible to duplicate the temperature at the 30-in. level within less than 0.5 F on successive tests, slight deviations appear between the two tabulated values for the temperatures at the 30-in. level obtained with the same radiator or convector. Since the eupatheoscope was less influenced by the presence of the cold exposed walls in position No. 3 than in position No. 1, position No. 3 was selected as the one on which to base all comparisons of the different heating units, although consistent comparisons could be made from the readings in position No. 1. The equivalent temperatures at the 30-in. level were all lower

than the 72.3 F corresponding to an effective temperature of 66 deg. Owing to the higher temperatures above this level, however, the general effect was one of comfort, and, in most cases, if a higher equivalent temperature was maintained at the 30-in. level the room felt stuffy to a person moving about in it.

For the tests shown in Fig. 6 the three cast-iron direct radiators maintained a temperature of approximately 68 F at the 30-in level, measured by means of a bare thermocouple made from No. 22 B. & S. gage wire. With the eupatheoscope in position No. 3 (Fig. 1), the equivalent temperature was practically the same as that observed with the thermocouple. The maximum difference of 0.3 F was obtained with the wall radiator and indicated that radiation was somewhat more effective in warming a large body than was reflected in the

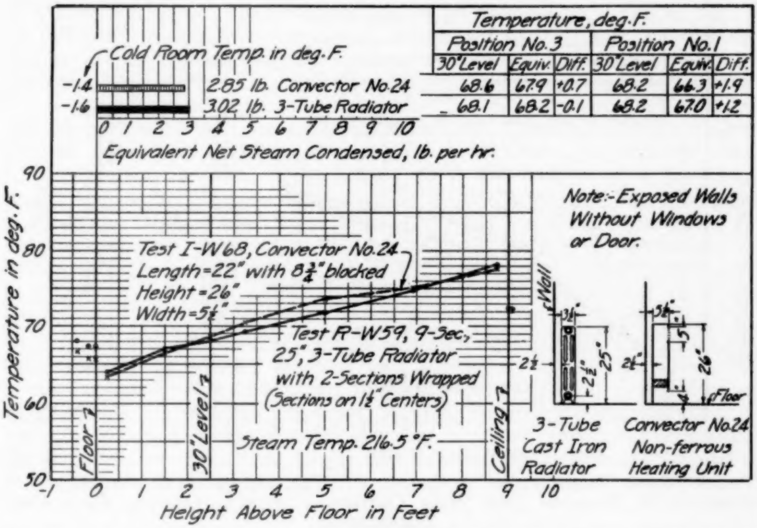

Fig. 8. Room Temperature Gradients and Equivalent Temperatures for a Convector and a Direct C. I. Radiator

thermocouple reading. With the eupatheoscope in position No. 1, the equivalent temperature was consistently lower than that shown by the thermocouple, thus proving that the closer proximity of the cold exposed wall had an appreciable effect on the degree of comfort. Just why this effect was greatest in the case of the wall radiator is not apparent. The steam condensation corresponded to the effectiveness in heating as reflected by the temperature gradients. That is, using the condensation for the 26-in., five-tube radiator as a base the reduction in condensate for the 26-in., three-tube radiator was 5.5 per cent and for the wall radiator was 13.9 per cent.

The results obtained with three types of convectors with 68 F maintained at the 30-in. level as compared with the 26-in. five-tube direct radiator under

the same conditions are shown in Fig. 7. It is evident from the eupatheoscope readings in position No. 3 that the convectors maintained an equivalent temperature approximately 1.5 F lower than that maintained by the direct radiator. This seems to indicate that some benefit results from direct radiation, particularly in the lower part of the room. It is also evident from the equivalent temperatures in position No. 1 that the proximity of the cold walls had more influence in the case of the convectors than in that of the direct radiator. The steam condensations are in the order that would be predicted from the temperature gradient curves. As compared with the direct radiator, convector

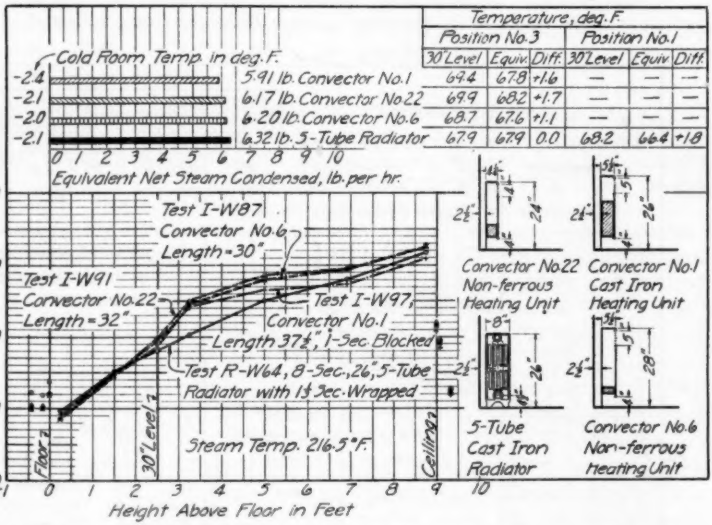

FIG. 9. ROOM TEMPERATURE GRADIENTS AND EQUIVALENT TEMPERATURES FOR A DIRECT C. I. RADIATOR AND THREE CONVECTORS WITH A COMMON EQUIVALENT TEMPERATURE

No. 6 showed a reduction in condensation of 3.1 per cent, convector No. 22 of 7.5 per cent, and convector No. 1 of 11.4 per cent.

The effect of cold doors and windows may be shown by comparing Figs. 7 and 8. For the tests shown in Fig. 8, the door and windows were covered with insulating board. With the eupatheoscope in position No. 3, elimination of the door and windows had very little effect in the case of the direct radiator, but in the case of the convector the difference between the equivalent temperature and the dry-bulb temperature at the 30-in. level was reduced from approximately 1.4 F to 0.7 F, or 50 per cent, based on the comparison of convectors Nos. 6 and 24, which were of the same type. With the eupatheoscope in position No. 1, the elimination of the door and windows in the case of the direct radiator, reduced the difference between the equivalent temperature and the dry-bulb temperature from 1.8 F to 1.2 F, or 33.4 per cent, and in the

case of the convector from 2.9 F to 1.9 F, or 34.5 per cent. The steam condensation cannot be compared because the units were of different size.

Since the results shown in Fig. 7 indicated that when the dry-bulb temperature at the 30-in. level was maintained at 68 F, the convectors produced an equivalent temperature less than that produced by the direct radiator, comparison of the steam condensations under those conditions might not give results truly representative of the relative heating effects. Hence, another series of tests were run for which the heat outputs of the different units were all adjusted to maintain the same equivalent temperature with the eupatheoscope in position No. 3. The equivalent temperature given by the direct radiator was selected as a standard, because in the previous tests with the direct radiator the equivalent temperature and the dry-bulb temperature at the 30-in. level were practically the same. The results of this series of tests are shown in Fig. 9. In this case, the dry-bulb temperature maintained at the 30-in. level by the convectors was approximately 1.5 F higher than 68 F. Expressed in terms of the steam condensation for the direct radiator, the condensation for convector No. 6 was reduced 1.9 per cent, for convector No. 22 was reduced 2.4 per cent, and for convector No. 1 was reduced 6.4 per cent. These reductions were less than the corresponding reductions of 3.1, 7.5, and 11.4 per cent obtained when a dry-bulb temperature of approximately 68 F was maintained at the 30-in. level. Hence, the conclusion expressed in a previous paper [1] that the steam condensation obtained when a temperature of 68 F is maintained at the 30-in. level is an inverse index of the heating effect in the case of convectors, is subject to slight modification in quantity but not in quality. Such a procedure would tend to credit the convectors with a heating effect somewhat too great. On the other hand, the difference is not so great as to invalidate the use of 68 F maintained at the 30-in. level as a simple and convenient criterion for testing convectors in order to obtain their relative worth from the standpoint of heating effect. The outstanding conclusion to be drawn from all of these tests, however, is that the maximum credit to be given to heating effect is probably in all cases less than 10 per cent, and the practice of increasing ratings anywhere from 10 to 30 per cent to allow for an extremely doubtful heating effect can hardly be justified.

Conclusions

The following conclusions may be drawn, governed by the conditions applicable to these tests:

1. The eupatheoscope affords a means of evaluating the combined effect of radiation and convection in a given environment in terms of a standard environment and in some terms related to human comfort.

2. As a provisional standard of comfort conforming to American practice, a room may be assumed to be comfortably heated when a sizable body at 83 F loses heat at the rate of 15.4 Btu per square foot per hour.

3. Future investigation should be undertaken to correlate the readings of the eupatheoscope with a more definite standard of comfort based on a consideration of all of the physical and physiological aspects of the problem.

4. The readings of the eupatheoscope serve as a means of ranking various heating units in the order of the relative heating effects produced.

5. The steam condensation obtained when the equivalent temperature is maintained at 68 F at the 30-in. level is a measure of the effectiveness of different types of heating units in providing for human comfort; lower condensations corresponding to greater effectiveness.

6. The steam condensation obtained when the dry-bulb temperature is maintained at 68 F at the 30-in. level is an approximate measure[1] of the relative effectiveness of different types of heating units in providing human comfort; lower condensation corresponding to greater effectiveness.

7. The heating effect is not materially less ~~greater~~ than the total heat output as measured by steam condensation under given standard conditions, and the practice of adding a large proportion to the condensation rating in order to provide for heating effect cannot be justified.

8. The proximity of cold walls has an appreciable effect on the degree of comfort as determined by the eupatheoscope.

DISCUSSION

C.-E. A. WINSLOW: The authors have rendered a very real service by this study of the new English device for measuring atmospheric influences. I note with much interest that they have fundamentally modified the instrument so that the eupatheoscope employed in their work is quite different from the instrument described by Dufton and his associates at Watford. We have worked with the British eupatheoscope in the Pierce Laboratory at New Haven and I feel sure that the instrument used by the authors is much better adapted to American problems.

The general principle involved marks, I think, a distinct advance. In the United States we have stressed air temperature, air movement and humidity and have almost wholly ignored radiation. The eupatheoscope measures the effect of air temperature, air movement and radiation, but ignores humidity. I am convinced that under most indoor conditions radiation is really a more important factor than humidity, and the fact that we can now allow for this factor is likely to prove illuminating.

H. F. HUTZEL: As a result of this investigation it appears that the eupatheoscope provides for a tangible means of measuring degree of comfort. I am not fully convinced, however, that the results obtained reflect the true degree of comfort.

The eupatheoscope is a comparatively new instrument and I do not feel qualified to intelligently criticize the manner in which it was applied in the test reported in this paper. There are several things, however, which are subject to criticism and of which, for my own information, I would like an explanation.

I can appreciate that the eupatheoscope affords a means of evaluating the probable comfort in terms of heat required to maintain the cylinder at a constant temperature, since the human comfort is directly correlated with the heat loss from the body. Whereas the human body may lose heat at the rate of 15.4 Btu per square foot per hour in an atmosphere of 72.3 deg, the percentage of this heat emitted by radiation

and convection will vary according to the surface temperature of the walls of the room—also, according to the type of clothes. For these experiments the eupatheoscope was provided with a blackened surface. I have questioned whether the blackened metal surface would lose or absorb the same percentage of heat by radiation and convection as the human body normally clothed. The authors have explained that the painted surface, clothes, the human skin, building materials, lamp black, rusty metal, in fact, nearly all surfaces except polished or semi-polished metal, radiate and absorb approximately 95 per cent of black body or total radiation and in view of the fact that they were dealing with low temperature or infra-red radiation, the type of finish would be of no consideration in this experiment. This explanation is not borne out by an experiment we have recently conducted. We applied 4-ply air cell pipe covering to a 10-ft length of 2-in. copper tubing placed in a horizontal position and determined the heat loss from this section by measuring the condensation. This section of tubing was then tested with a copper sheet of mill finish wrapped around the same 4-ply air cell pipe covering and it was found by measuring the condensation that the heat loss was 10.9 per cent less. Since we were likewise dealing with a low temperature range, I am led to believe that the tests with the eupatheoscope may have been in error.

During the course of these tests, the outside atmosphere surrounding the cold room was held at a temperature of 2 deg below zero. It is fair to assume that the surface temperature on the inside of this test room was in the neighborhood of 58 deg. If this is true it is also fair to assume that the blackened body would radiate more heat than if this body had a different type of surface. Does the human body normally clothed radiate the same amount of heat as the blackened metal surface when subject to the same temperature difference? It would seem to me that, in order to obtain correct results, the eupatheoscope should have a cloth surface similar to that worn by the human body and that this eupatheoscope should be calibrated with such a surface instead of with a blackened surface. The results of tests might then be different, for this report shows an appreciable difference in results when the eupatheoscope is located in different positions in the room. If the cloth surface radiates less heat than the blackened metal surface in the tests made with the convector type of heater, the equivalent temperatures would be proportionately higher.

Although this report does not indicate whether or not the test room was furnished, I presume that it was not, for the photograph shows no furniture other than some curtains hanging at the window. Had this room been furnished as an average room in a house, it appears to me that the effect of radiation might have been different. This, too, might have had a different bearing on the conclusions drawn.

I can conceive of this eupatheoscope absorbing considerable radiant energy from the radiator in a room temperature which would be too low for human comfort. I cite the case of the effect of a fireplace in a room in which the atmosphere is cool. If one faces the fire his back feels cold. One feels uncomfortable under these conditions even though the rate of heat absorption from the fireplace may be far in excess of that dissipated by the human body.

The point I am trying to bring out is that the eupatheoscope in the room heated by a direct radiator probably absorbs more heat on the side facing the radiator than on the shaded side. In this case the equivalent temperature indicated might not reflect the comfort temperature. The convector, on the other hand, does not radiate much heat energy. It is probable that in this case the eupatheoscope gave off heat

at a uniform rate throughout its surface. Accordingly, even though the results indicate a different effect of temperature between the direct and convector type of heaters, the true degree of comfort might have been the same or even better in the case of a convector.

No. 963

TESTS OF CONVECTORS IN A WARM WALL TESTING BOOTH (Part II)

By A. P. KRATZ,† M. K. FAHNESTOCK ‡ (*MEMBERS*) AND E. L. BRODERICK § (*NON-MEMBER*)

URBANA, ILL.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and conducted at the University of Illinois

The data presented in this paper were obtained in connection with an investigation conducted by the Engineering Experiment Station of the University of Illinois, of which Dean M. S. Ketchum is the Director. The work is carried on in the Department of Mechanical Engineering under the direction of A. C. Willard, Professor of Heating and Ventilation, and Head of the Department. This paper includes results from the part of the work constituting a continuation of the program devoted to the study of the performance of convectors in a warm wall booth and the material will ultimately comprise part of a bulletin of the Engineering Experiment Station.

A PREVIOUS paper [1] on this subject was devoted to a discussion of tests run in a warm wall testing booth for the purpose of deciding the validity of the correction factor, applied in the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS' Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code) [2] for reducing the heat output obtained under test conditions to the equivalent heat output under standard conditions with steam in the heating unit at 215 F and a temperature of 65 F for the air at the inlet. This correction factor is:

$$C = \left[\frac{150}{t_s - t_1}\right]^{1.8}$$

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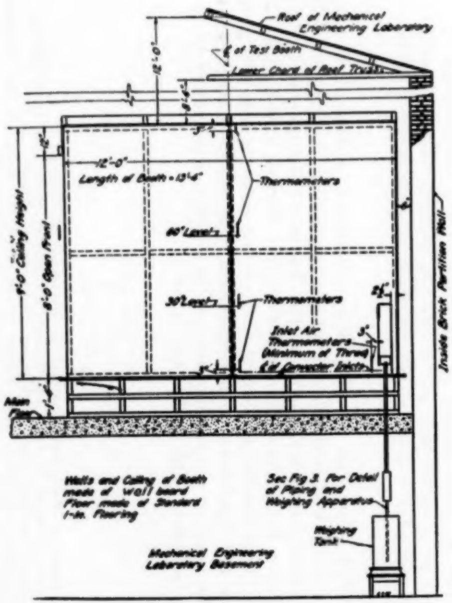
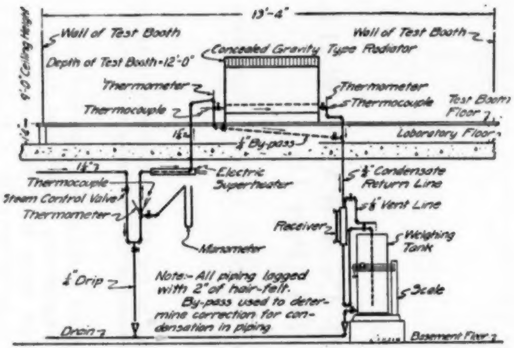
‡ Special Research Assistant Professor in Mechanical Engineering, University of Illinois.
§ Research Assistant in Mechanical Engineering, University of Illinois.
[1] Tests of Convectors in a Warm Wall Testing Booth, by A. P. Kratz, M. K. Fahnestock, and E. L. Broderick, A. S. H. V. E. TRANSACTIONS, Vol. 38, 1932, p. 511.
[2] A. S. H. V. E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code). A. S. H. V. E. TRANSACTIONS, Vol. 37, 1931, p. 367.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Hotel Statler, Detroit, Mich., June, 1933, by A. P. Kratz.



A—Section of warm wall test booth



B—Detail of piping for warm wall test booth

FIG. 1. WARM WALL TEST BOOTH USED IN STUDY OF PERFORMANCE OF CONVECTORS

where 150 = the temperature difference between steam at 215 F and inlet air at 65 F.
t_s = the temperature of steam during the test.
t_i = the inlet air temperature during the test.

This correction factor is the one commonly used in the case of cast-iron direct radiators and its validity has been well established when used in this connection, but its validity had not been well established when used in connection with convectors.

The results [1] in general indicated a close agreement between the heat output actually obtained with steam at 215 F and inlet air at 65 F, and that calculated from the heat output at some other inlet air temperature through the medium of the correction factor, provided that the inlet air temperature did not exceed 80 F or fall below 60 F. There seemed to be some indication, however, that the percentage deviation between the actual and calculated heat outputs might be affected by the physical dimensions of the convectors, particularly by the height of the cabinet. The tests were, therefore, continued for the purpose of determining to what extent the dimensions of the convectors influence the limits of accuracy in the application of the correction factor.

DESCRIPTION OF APPARATUS

A section of the test booth and a detail of the piping are shown in Fig. 1 *A* and *B*. This has been fully described both in a previous paper [1] and in the A. S. H. V. E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code).[2]

Four types of convectors were tested. Two of these types were tested with three different heights of cabinets and one type was tested with two different heights of cabinets. Three types were tested with different widths of heating units and three with different lengths. The types and dimensions of all convectors are shown in Table 1 and in the insets in Figs. 2 to 7.

TEST PROCEDURE

All tests were run with a steam temperature of 216.5 F in the convectors. In order to obtain curves for each convector, establishing the relation between the heat output, as measured by steam condensation, and the temperature of the air at inlet, tests were run at different inlet air temperatures, varying over a range of 60 F to 90 F. In order to accomplish this, the large room in which the test booth was erected was heated or cooled to a temperature approximating the desired inlet air temperature, and the convector was allowed to establish the temperature conditions in the booth necessary for thermal equilibrium.

No test observations were made until conditions had remained constant for several hours, as indicated by readings of all thermocouples and thermometers. When the required thermal constancy had been attained, the condensate was weighed over the period of one hour, and no test was accepted if the condensate showed more than 2½ per cent deviation in the successive 10-min increments of weight. At the end of each test, a separate test was run to

TABLE 1. DEVIATION OF CALCULATED CORRECTION FROM TEST CURVE

Fig. No.	Con-vector No.	Type of Heating Unit	Overall Dimensions of Convector, Inches			Height of Heating Unit, Inches	60 F Inlet				80 F Inlet			
							Heat Output Btu per Hour		100 x Ratio $\frac{c^{n}155}{a^{n}155}$	Per Cent Difference	Heat Output Btu per Hour		100 x Ratio $\frac{c^{n}135}{a^{n}135}$	Per Cent Difference
			Length	Height	Width		By Test $a^{n}155$	Calcu-lated $c^{n}155$			By Test $a^{n}135$	Calcu-lated $c^{n}135$		
4	* 5	Non-ferrous	36	18	5½	2	5575	5515	98.9	−1.1	4410	4605	104.4	+4.4
4	* 6	Non-ferrous	30	28	5½	2	6225	6195	99.6	−0.4	5065	5175	102.2	+2.2
4	17	Non-ferrous	26	66	5½	2	6890	6885	99.9	−0.1	5725	5755	100.5	+0.5
2	19	Non-ferrous	29	18	5⅜	5	3895	3900	100.1	+0.1	3260	3255	99.9	−0.1
2	11	Non-ferrous	29	29	5⅜	5	4975	4970	99.9	−0.1	4115	4150	100.8	+0.8
2	16	Non-ferrous	29	67	5⅜	5	6680	6670	99.9	−0.1	5510	5570	101.0	+1.0
3	* 1	Cast Iron	37½	26	5½	14³⁄₁₆	6270	6275	100.1	+0.1	5260	5245	99.7	−0.3
3	*13	Cast Iron	37½	26	4⅛	14³⁄₁₆	5680	5670	99.8	−0.2	4700	4740	100.8	+0.8
3	18	Cast Iron	38	70	5¾	14³⁄₁₆	8270	8230	99.5	−0.5	6775	6880	101.6	+1.6
6	23	Non-ferrous	30	28½	10⅞	2	10470	10450	99.8	−0.2	8655	8730	100.9	+0.9
5	21	Non-ferrous	20	24	4¼	4	3695	3655	98.9	−1.1	2910	3050	104.8	+4.8
5	22	Non-ferrous	32	24	4¼	4	6125	6060	99.0	−1.0	4870	5070	104.1	+4.1
7	20	Non-ferrous	21½	29	11	5	5390	5350	99.3	−0.7	4315	4465	103.5	+3.5
—	* 2	Non-ferrous	31	30⅜	5	6	5830	5810	99.7	−0.3	4820	4860	100.8	+0.8
—	* 3	Non-ferrous	21	30⅜	7½	6	5200	5185	99.7	−0.3	4310	4340	100.7	+0.7
—	* 7	Bimetallic	22⅝	28⅜	6¹⁄₁₆	20	6780	6805	100.4	+0.4	5740	5695	99.2	−0.8

* Presented in previous paper, Tests of Convectors in a Warm Wall Testing Booth, by A. P. Kratz, M. K. Fahnestock, and E. L. Broderick, A. S. H. V. E. TRANSACTIONS, Vol. 38, 1932, p. 511.

determine the condensation in the piping alone, and the total condensate was corrected by subtracting the amount so determined.

RESULTS OF TESTS

The results of the tests are shown as full line curves in Figs. 2 to 7. These curves indicate the mean trend of the points representing the individual tests. The differences in temperature between the steam in the heating unit and the air at the inlet of the enclosure have been used as an abscissæ for all curves, but since the tests were all run with a steam temperature of 216.5 F, the abscissæ are also representative of the temperature of the air at inlet, and the latter may be obtained by subtracting any given temperature difference from 216.5 F.

In order to determine the probable deviation of the corrected heat outputs, resulting from the use of the correction factor as provided for in the A. S. H. V. E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code) [2] from the values that would have been obtained by actual tests at various temperature ranges between steam and inlet air, the calculated curves, shown as broken lines, have been superimposed on the test curves in Figs. 2 to 7. These calculated curves are defined by the equation:

$$H = a^{H}150\left[\frac{t_s - t_i}{150}\right]^{1.8}$$

where H = the calculated heat output at any temperature difference between steam and inlet air, Btu per hour.
$a^{H}150$ = the actual heat output read from the test curve at a temperature difference of 150 F, Btu per hour.
t_s = the steam temperature for which H is to be calculated, deg F.
t_i = the inlet air temperature for which H is to be calculated.

Since the standard temperature difference was defined as from 215 F to 65 F, or 150 F, both the actual and the calculated curves have been made coincident at the heat output corresponding to a temperature difference of 150 F, and the difference between these curves at any given temperature range represents the error in the application of the correction factor to test results obtained with that temperature difference. In all cases the deviation, or per cent error, in the application of the correction factor at the lower limiting value of the temperature of the inlet air, or 60 F, as specified in the A. S. H. V. E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code),[2] was negligible, and the discussion has been confined to the higher limiting value or 80 F. The latter corresponds to a temperature difference of 135 F.

The effect of varying the height of the cabinet is shown in Figs. 2, 3, and 4. For Fig. 2, tests were run on heating units having the same length and width with three different heights of cabinets, and for Fig. 3 on heating units having the same length and width with two different heights of cabinets. For these two types of convectors it is apparent that as the height of cabinet was increased the deviation between the actual and calculated heat outputs for a temperature range of 135 F, or the error in using the correction factor also increased. For Fig. 4 tests were run with three different heights of cabinets on the same type of convector having heating units of the same width but of

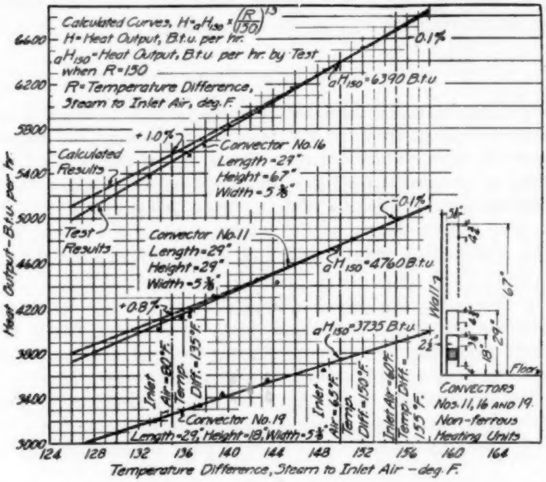


FIG. 2. PERFORMANCE CURVES FOR CONVECTORS NOS. 11, 16, AND 19 SHOWING EFFECT OF HEIGHT

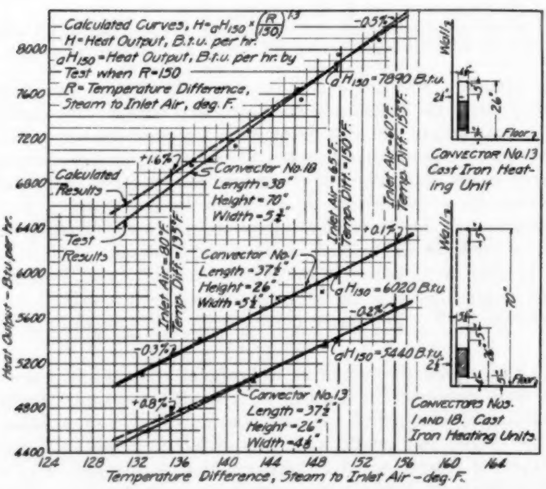


FIG. 3. PERFORMANCE CURVES FOR CONVECTORS NOS. 1, 13, AND 18 SHOWING EFFECT OF HEIGHT

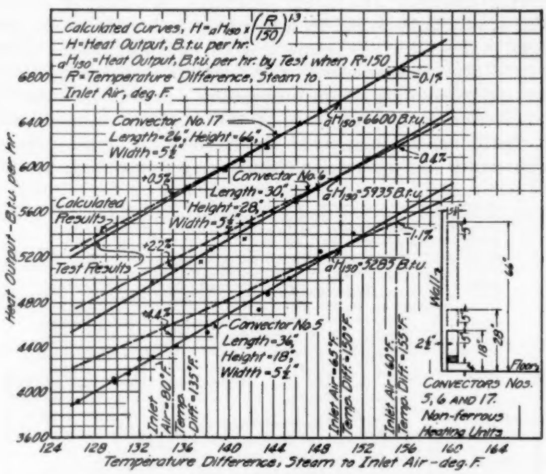


FIG. 4. PERFORMANCE CURVES FOR CONVECTORS NOS. 5, 6, AND 17 SHOWING EFFECT OF HEIGHT

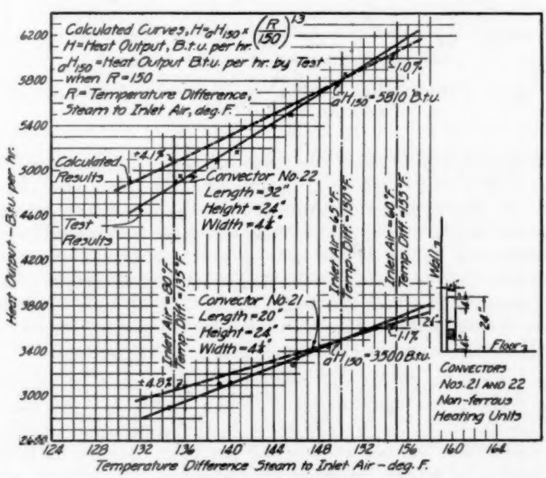


FIG. 5. PERFORMANCE CURVES FOR CONVECTORS NOS. 21 AND 22 SHOWING EFFECT OF LENGTH

slightly different lengths. The effect of length, however, is small, as shown in Fig. 5, and if anything would tend to increase the deviations shown by the lower cabinets in Fig. 4 if the same lengths had been used. Hence, results shown in Fig. 4 are representative of the nature of the effect of height and approximately representative of the magnitude. In this case the error in the use of the correction factor decreased as the height of cabinet was increased. Apparently, therefore, the height of the cabinet has some influence on the accuracy with which the correction factor may be applied, but the type of

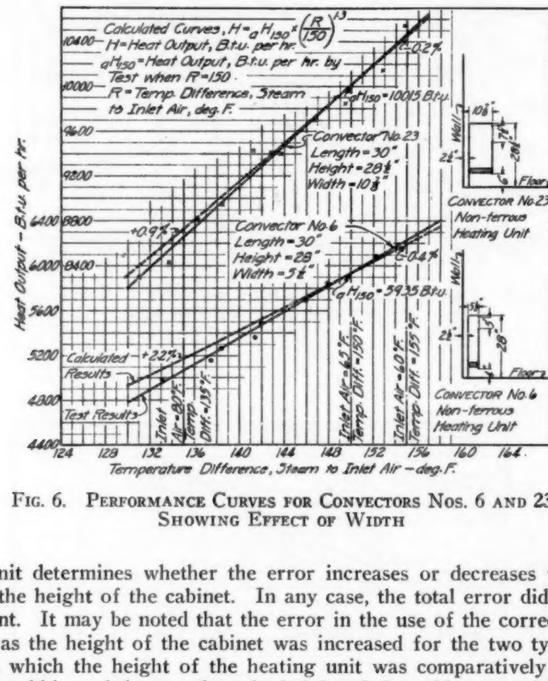


FIG. 6. PERFORMANCE CURVES FOR CONVECTORS NOS. 6 AND 23 SHOWING EFFECT OF WIDTH

heating unit determines whether the error increases or decreases with an increase in the height of the cabinet. In any case, the total error did not exceed 4.5 per cent. It may be noted that the error in the use of the correction factor increased as the height of the cabinet was increased for the two types of convectors in which the height of the heating unit was comparatively large relative to the width, and decreased as the height of the cabinet was increased for the type of convector in which the height of the heating unit was comparatively small relative to the width. This fact may be significant, but the amount of data is hardly sufficient to warrant drawing a general conclusion.

The effect of the length of the convector for cabinets of the same height and heating units of the same width is shown in Fig. 5. As the length was increased from 20 in. to 32 in., the error in the application of the correction factor decreased from 4.8 per cent to 4.1 per cent. Hence, it is evident that the length had very slight influence on the error. This type of convector showed the greatest uncertainty in the application of the correction factor of all the convectors tested. The maximum error, however, did not exceed 5 per cent.

The effect of the width of the heating unit is shown in the two lower curves in Fig. 3 and in Figs. 6 and 7. For the types of convectors shown in Figs. 3 and 6 the error in the application of the correction factor decreased as the width of the heating unit was increased, while for the type shown in Fig. 7 the reverse was true. The width apparently has some influence on the error, but the nature of the influence is dependent on the type of convector.

The error in the application of the correction factor for all of the types of convectors tested and reported on in this paper and the previous paper,[1] for both 60 F and 80 F inlet air temperatures is shown in Table 1. In the majority of cases the error was of the order of one per cent. This error was influenced

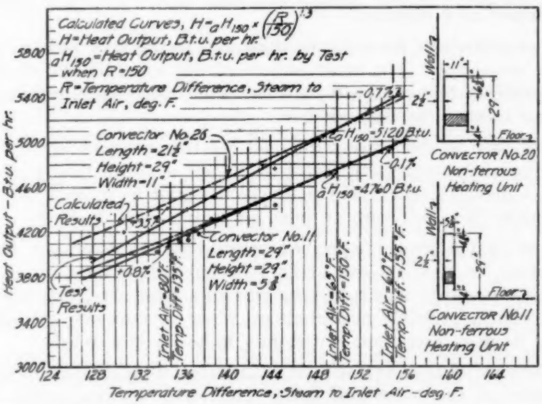


FIG. 7. PERFORMANCE CURVES FOR CONVECTORS NOS. 11 AND 20 SHOWING EFFECT OF WIDTH

by both the type and the physical dimensions of the convector, and the exact nature of the influence was uncertain and difficult to predict. The error did not exceed 5 per cent, however, for any of the convectors tested.

In the case of tests run with an inlet air temperature of 80 F, or the upper limit of tolerance specified in the Code, the curves show that the actual application of the correction factor to the test results would, in practically every case, have given a calculated equivalent heat output, or rating, less than the actual heat output that would have been obtained if the test had been run with steam in the radiator at 215 F and the inlet air temperature at 65 F. That is, the correction would result in safe rating for the convector. If this upper limit of tolerance for the inlet air temperature were reduced to 75 F, the maximum probable error in the application of the correction factor would be reduced to approximately 3 per cent, instead of the probable maximum of 5 per cent existing with the present upper limit of tolerance of 80 F.

In the case of tests run with an inlet air temperature of 60 F, or the lower limit of tolerance, the actual application of the correction factor to the test results would result in a rating slightly higher than the actual performance of the convector with steam at 215 F and inlet air at 65 F. Since the maximum

error would not exceed approximately one per cent, however, this slight overrating would not be serious, and the lower limit of tolerance may be regarded as satisfactory.

CONCLUSIONS

The following conclusions may be drawn from all of the test results, and they supersede those given in the previous paper[1]:

(1) The error in the application of the correction factor specified in the A. S. H. V. E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code),[2] probably does not exceed 5 per cent when the temperature of the inlet air on the test is not more than 15 F above or 5 F below the 65 F adopted as a standard.

(2) The magnitude of the error is influenced by the type and physical dimensions of the convector. The exact nature of this error is uncertain, and it is difficult to predict whether it will increase or decrease when the dimensions of the convector increase or decrease.

(3) In the case of tests run with inlet air temperatures higher than 65 F, the application of the correction factor to the test data practically always results in a rating somewhat lower than that which would have been obtained if the test had actually been run with an inlet air temperature of 65 F.

(4) Lowering the present specified upper limit of tolerance for the inlet air temperature, or 80 F, to one of 75 F would reduce the error in the application of the correction factor to tests run with inlet air temperatures greater than 65 F from a probable maximum of 5 per cent to one of approximately 3 per cent. Such a restriction is certainly to be recommended.

(5) In the case of tests run with inlet air temperatures between 60 and 65 F, the application of the correction factor practically always results in a rating slightly higher than the actual performance of the convector. Such overrating does not exceed approximately one per cent, and the lower limit of tolerance of 60 F, as now specified, may be regarded as satisfactory.

No. 964

THE HEAT CONDUCTIVITY OF WOOD AT CLIMATIC TEMPERATURE DIFFERENCES

By F. B. Rowley,† (*MEMBER*), Minneapolis, Minn.

This paper is the result of research sponsored by the America Society of Heating and Ventilating Engineers in cooperation with the *National Lumber Manufacturers' Association* and conducted at the University of Minnesota

THE thermal properties of wood are of importance both in building construction to conserve fuel and improve living conditions, and in the field of refrigeration. Little information on the subject has previously been available, and the few tests on which previous fragmentary data have been based have omitted to record sufficient data on the character and condition of material tested so that results could be interpreted in terms of general utility or extended to apply to species not tested.

In 1929, therefore, a preliminary study was made to determine the feasibility of a more extended research program and to identify the factors which might require investigation.

Based on the results of this preliminary program, the project was extended to cover about thirty species of wood and to determine the effect of density and moisture content on the thermal conductivity of several representative species. The work has been a cooperative project between the University of Minnesota, the American Society of Heating and Ventilating Engineers, and the *National Lumber Manufacturers' Association*.

The thermal properties of wood as used in practice are influenced by a number of factors, of which the more important are species, density, moisture content, direction of heat flow, inclination of grain, and relation of volume or thickness to moisture content.

Preliminary Investigation

The work in 1929 included over 100 tests of selected panels from 14 species, the material being selected by the U. S. Forest Products Laboratory as representative in character. Study of the results led to the following tentative conclusions:

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1. That conductivity varies approximately in a straight line ratio with density in any given species.

2. That somewhat greater conductivity may be expected tangential to grain (*i. e.,* in edge-grain material) than radially, in species with strongly marked annual rings.

3. That no consideration need be given to the position of annual rings in species of uniform grain.

4. That small crevices such as occur between boards in ordinary construction do not materially affect conductivity over the area tested, though they may, in the absence of suitable precautions, affect air infiltration.

5. That conductivity varies substantially in a straight line ratio with moisture content if the latter is expressed in terms of weight per volume of wood.

6. That the inherent variability of wood makes a considerable number of tests of each species necessary in order to arrive at a fairly representative value for each species.

FULL SCALE INVESTIGATION

Pursuant to the above findings, the following commercial species were selected for investigation:

California Redwood
Douglas Fir
Eastern Hemlock
Hard Maple
Longleaf Yellow Pine
Norway Pine
Ponderosa Pine
Red Cypress
Red Oak
Shortleaf Yellow Pine
Sitka Spruce
Soft Elm
Soft Maple
Sugar Pine
Western Red Cedar
West Coast Hemlock
Western Larch
White Ash
White Fir
White Oak
Northern White Pine
Yellow Birch

Samples of each of these species were selected covering the range of density normally experienced in that species, and tested to determine the relation between thermal conductivity and density and the average conductivity for the species. In order to reduce the volume of test work for a complete determination of the moisture-conductivity relation, they were divided into six groups and a single species from each group was tested. The groups and representatives selected for test were as follows:

SPECIES IN GROUP	REPRESENTATIVE SPECIES TESTED
Norway Pine Sugar Pine Northern White Pine Ponderosa Pine	Ponderosa Pine
California Redwood Western Red Cedar Red Cypress	Red Cypress
Douglas Fir Longleaf Yellow Pine Western Larch Shortleaf Yellow Pine	Shortleaf Yellow Pine

Species in Group	Representative Species Tested
Eastern Hemlock Sitka Spruce White Fir West Coast Hemlock	West Coast Hemlock
White Ash White Oak Red Oak	White Oak and Red Oak
Hard Maple Soft Elm Soft Maple Yellow Birch	Yellow Birch

For the six species selected as representatives of the various groups, a number of tests were made at various moisture contents to determine the moisture-thermal conductivity relation. The curve obtained from each of these samples was taken as a typical moisture-conductivity relation for each of the other species in the group. While this procedure leaves some doubt as to whether the results for the species tested apply accurately for the others in the group represented, the similarity of cell structure and other pertinent characteristics makes it probable that any departures involved are less than might be experienced in any one sample of the species actually tested.

Selection of Samples

The samples were selected by the *National Lumber Manufacturers' Association* and conditioned to approximately 8 per cent of moisture. These samples were sent to the University of Minnesota where they were selected in order to get uniform test panels of different densities. In this selection, boards of substantially the same densities were used to make up a pair of test panels, and panels were made up covering as wide a range of densities as possible from each species.

When the tests were first started, test samples 24 in. square were prepared for testing in the 24-in. hot plate. It later developed that the results obtained from the 12-in. hot plate were equally dependable and that it was possible to select much more uniform samples of this size. The majority of the tests, therefore, were run on 12-in. square panels. Those samples, a part of which were tested by the 24-in. hot plate, were: Douglas Fir, Western Red Cedar, White Fir, and Red Cypress. These samples are indicated in Table 2 by asterisks.

Preparation of Samples

In preparing the test panels, the boards for a complete set were run through a planer in order to give a smooth surface on both sides and to make sure that all of the boards for each sample were of uniform thickness. The boards were then weighed and selected for density. A sample was taken from each board for moisture determination; the average of the results for all boards of a pair of test panels was taken as the average for the test.

The pieces used for moisture determination were wrapped in oiled paper as soon as cut from the test specimen. They were then taken to the Laboratory,

weighed, and dried to a constant weight at 175 F. The loss of weight divided by the dry weight was taken as the percentage of moisture in the sample.

In some of the earlier tests, the moisture determinations were made at 212 F. These species were: Douglas Fir, White Fir, and Western Cedar. All others were dried at 175 F. In the case of Red Oak, the moisture content samples were dried at both 175 F and 212 F, which resulted in an increase of about 1 per cent in loss for the higher temperature.

For some of the samples, there was a slight variation in the moisture con-

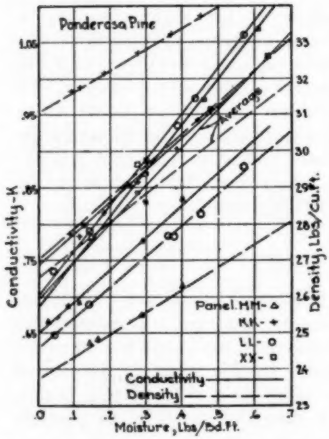


FIG. 1. PONDEROSA PINE—THE EFFECT OF MOISTURE ON DENSITY AND CONDUCTIVITY

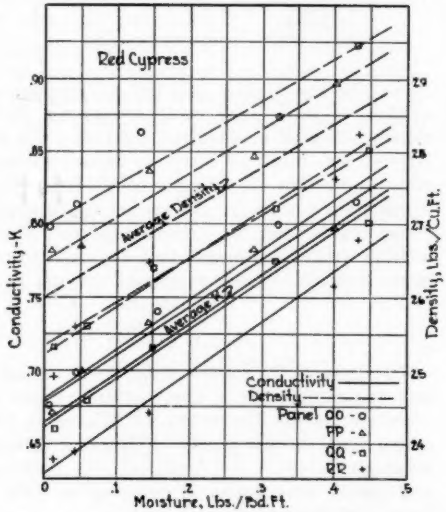


FIG. 2. RED CYPRESS—THE EFFECT OF MOISTURE ON DENSITY AND CONDUCTIVITY

tent during the test, which was determined by weighing the panels both before and after testing. In these cases, the average moisture content was taken as the moisture content for the particular test, although the variation was so slight in most cases that it was not necessary to take averages. An investigation was made to determine the effect of this moisture change during the test on thermal conductivity as obtained by the hot plate. It was shown that even though all of the heat for evaporating the moisture might come from that supplied to the test section the error would be less than 1 per cent and in all cases negligible.

When the moisture content of a panel is given as the percentage by weight, the test results may be misleading, due to the fact that for the same percentage by weight there will be different amounts of moisture per unit volume. For this reason, the percentage of moisture by weight was converted to pounds of moisture per board foot. In making this conversion, the volume of the sample was determined by measuring the length and width and by taking the

thickness as determined in the test plate. The formula used for making the conversion was as follows:

$$\text{Weight of moisture per board foot (144 cu in)} = \frac{\text{(Average moisture per cent before and after test, dry basis)} \times \text{(dry wt sample)} \times 144}{(2 \times \text{thickness of 1 panel}) \times \text{length} \times \text{width}}$$

DENSITY

Since, in some cases, there might be a slight change in weight during the test, the density calculations were all made on the basis of weight and thickness taken at the beginning of the test, the change in density for a slight change in moisture being negligible. The warping of wood which sometimes occurs during a test might give a greater thickness reading of the test panel at the end of the test than at the beginning, even though no additional moisture was added. In some cases, there was a slight warping of the test boards before the test, making it necessary to take the thickness readings of the individual boards and not to use the thickness as obtained during the test. When there was any appreciable warping, additional samples were selected and tested. In all cases of tests upon which the present data have been based, the density has been determined by the following formula:

$$\text{Density, pounds per cubic foot before test} = \frac{1728 \times \text{weight before test}}{(2 \times \text{thickness of 1 panel}) \times \text{length} \times \text{width (before test)}}$$

METHOD OF TESTS

All thermal conductivity tests were made with standard hot plate apparatus, and in practically all of the tests, the 12-in. square plates were used. In each case, the test samples were prepared and the panels were placed in the test plate and left at constant conditions a sufficient length of time to insure uniform temperature gradients throughout the specimen. The temperatures of the test surfaces were then taken and the rate of heat flow through the test section was calculated from the electric input to the heating element. Readings at constant conditions were continued for average periods of five hours in order to insure uniform data. The thickness of the sample from which the thermal conductivities were calculated was taken as the distance between the test plates during the test.

DENSITY-MOISTURE AND CONDUCTIVITY-MOISTURE RELATIONS

As previously stated, all of the species of wood were divided into six groups, and a representative species from each group was selected for determining the moisture-conductivity relation. From each of these representative species, four or more pairs of test panels were selected, and the conductivity and density of each pair were determined at several moisture contents covering the range from 0 to approximately the point of fiber saturation for the particular sample. In the earlier tests, a much wider range of moisture percentage was used, but for the later tests, those points above the fiber saturation were eliminated. The moisture contents of the samples were changed by placing them in a saturated atmosphere for a sufficient length of time to take up the required moisture. They were then wrapped in oiled paper and left for periods of two weeks

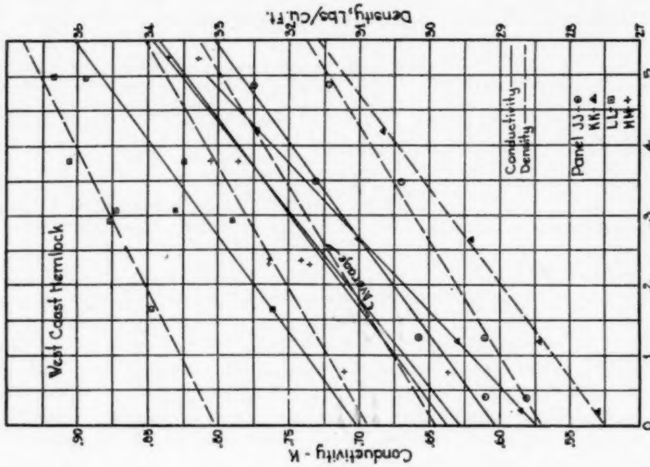


FIG. 4. WEST COAST HEMLOCK—THE EFFECT OF MOISTURE ON DENSITY AND CONDUCTIVITY

or longer to give the moisture an opportunity to equalize throughout the samples.

In the denser wood, such as oak and birch, the higher moisture contents were accompanied by excessive warping of the boards. The condition was so bad in the oak group that it was necessary to run a large number of panels of both white oak and red oak in order to obtain satisfactory curves, two curves being taken from each group and averaged to give the final correction curve. No particular difficulty was experienced in conditioning the lighter species,

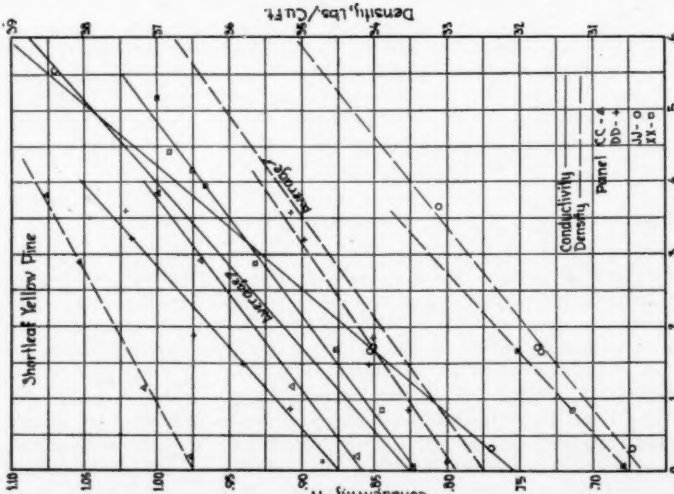


FIG. 3. SHORTLEAF YELLOW PINE—THE EFFECT OF MOISTURE ON DENSITY AND CONDUCTIVITY

and with these there was but little trouble experienced in warping during the tests.

The slopes of the several curves of each species were averaged to give the

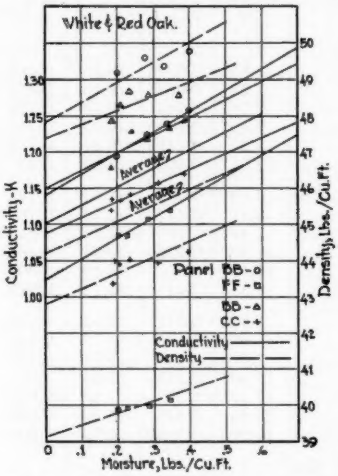


FIG. 5. WHITE AND RED OAK—THE EFFECT OF MOISTURE ON DENSITY AND CONDUCTIVITY

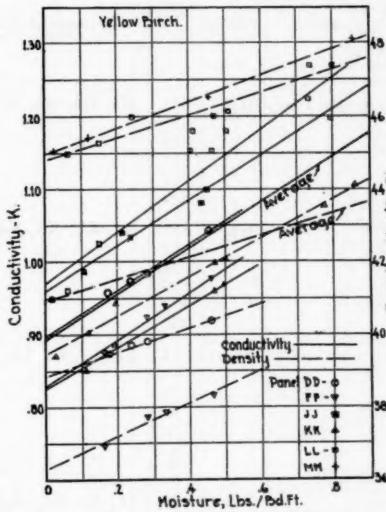


FIG. 6. YELLOW BIRCH—THE EFFECT OF MOISTURE ON DENSITY AND CONDUCTIVITY

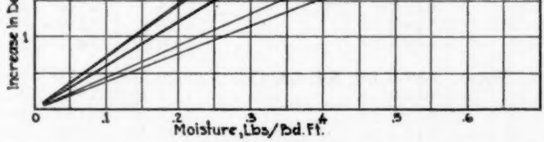


FIG. 7. THE RELATION BETWEEN DENSITY AND MOISTURE CONTENT FOR THE REPRESENTATIVE SPECIES AS INDICATED

slope of the correction curve. This was done for both the moisture-conductivity and the density-moisture relation. The test results for the moisture-conductivity and the density-moisture relation are shown in the tabulated data of Table 1 and in the curve sheets, Figs. 1 to 6, inclusive.

TABLE 1. TEST DATA FOR MOISTURE-DENSITY AND MOISTURE-CONDUCTIVITY CORRECTION CURVES

Species	Panels	Test No.	Composed of Boards	Moisture Per Cent by Weight Dry Basis	Moisture Lb/Bd Ft as Tested	Density Lb/Cu Ft as Tested	Conductivity "K" 75 Deg M. T.
Ponderosa Pine.	HH′	881	H3, H4, H5, I3.....	1.85	0.036	23.86	0.665
		877	....	3.47	0.067	24.10	0.679
		873	....	7.54	0.144	24.70	0.713
		545	....	8.73	0.166	24.83	0.718
		891	....	15.83	0.289	25.47	0.774
		899	....	22.65	0.401	26.29	0.836
	KK′	865	K1, K2, K3, K4, K5.	3.62	0.092	31.64	0.792
		882	....	4.41	0.112	31.72	0.794
		864	....	7.24	0.181	32.14	0.815
		889	....	11.30	0.276	32.67	0.861
		902	....	15.48	0.370	33.19	0.901
		900	....	18.85	0.444	33.71	0.945
	LL′	733	L1, L2, L3, L4......	1.99	0.041	24.90	0.728
		729	....	6.68	0.135	25.77	0.787
		756	....	7.01	0.141	25.74	0.782
		723	....	18.40	0.355	27.65	.....
		770	....	19.37	0.372	27.63	0.927
		762	....	23.65	0.448	28.23	0.984
		759	....	29.69	0.563	29.54	1.058
	XX′	440	J1, J3, K1, K2, K3, K4, L1, L2, L3, L4	5.78	0.133	.....	0.803
		781	....	12.55	0.275	29.61	0.865
		752	....	13.67	0.294	29.34	0.861
		746	....	13.94	0.301	29.74	0.853
		767	....	22.54	0.476	31.14	0.986
		764	....	29.88	0.631	32.61	1.083
Red Cypress...	OO′	656	L1, M3, M4, K1....	0.29	0.007	26.95	0.675
		642	....	1.95	0.044	27.26	0.698
		620	....	7.08	0.156	28.25	0.740
		711	....	15.75	0.322	28.47	0.800
		689	....	21.33	0.430	29.44	0.816
	PP′	657	K2, L2, M1, M2....	0.51	0.012	26.62	0.671
		641	....	2.36	0.052	26.70	0.700
		621	....	6.67	0.144	27.74	0.733
		710	....	14.22	0.289	27.92	0.783
		688	....	20.02	0.401	28.94	0.798
	QQ′	658	K3, K4, L3, J5......	0.71	0.015	25.31	0.660
		640	....	2.78	0.058	25.62	0.679
		623	....	7.29	0.150	26.40	0.716
		713	....	16.38	0.318	27.20	0.775
		687	....	22.16	0.446	28.04	0.802
	RR′	655	L4, J2, J4, L5.......	0.60	0.013	24.90	0.639
		643	....	2.04	0.043	25.58	0.645
		619	....	7.08	0.145	26.47	0.671
		712	....	20.82	0.401	27.62	0.758
		690	....	22.81	0.433	28.23	0.789

NOTE.—In cases where the conductivity or density is omitted from the test results, the warping of the samples made the results of doubtful value.

TABLE 1. TEST DATA FOR MOISTURE-DENSITY AND MOISTURE-CONDUCTIVITY CORRECTION CURVES—*Continued*

Species	Panels	Test No.	Composed of Boards	Moisture Per Cent by Weight Dry Basis	Moisture Lb/Bd Ft as Tested	Density Lb/Cu Ft as Tested	Conductivity "K" 75 Deg M. T.
Shortleaf Yellow Pine..	CC′	883	B2, C2............	0.06	0.020	36.52	0.862
		879	....	3.89	0.116	37.17	0.907
		893	....	10.05	0.289	38.06	0.969
		905	....	13.54	0.382	38.53	0.998
	DD′	884	D3, D1...........	0.47	0.013	33.01	0.886
		880	....	3.13	0.085	33.52	0.908
		876	....	5.45	0.147	34.08	0.940
		434	....	6.95	0.185	34.01	0.974
		906	....	12.34	0.319	34.95	1.017
		908	....	13.91	0.357	35.14	1.021
	JJ′	732	J2, J3, K3, L3......	1.23	0.031	30.45	0.770
		728	....	6.65	0.165	31.70	0.853
		757	....	6.85	0.170	31.76	0.851
		722	....	15.31	0.366	33.10	....
		770	....	24.11	0.552	.....	1.070
		759	....	29.31	0.671	35.66	1.153
	XX′	453	J1, J3, K1, K2, L2, L3...............	0.21	0.006	30.58	0.824
		438	....	3.33	0.084	31.28	0.845
		429	....	6.75	0.167	32.03	0.877
		782	....	11.99	0.284	31.79	0.932
		753	....	16.90	0.392	32.60	0.966
		747	....	17.78	0.413	32.95	0.976
		768	....	19.15	0.438	32.78	0.992
		764	....	22.87	0.514	33.25	0.999
West Coast Hemlock....	JJ′	668	J2, K5, L4, L2......	1.72	0.041	28.60	0.610
		666	....	5.38	0.125	29.20	0.657
		726	....	16.14	0.348	30.40	0.730
		714	....	23.15	0.484	31.43	0.774
	KK′	670	K5, K4, J3, K3.....	0.93	0.021	27.60	0.585
		664	....	5.30	0.119	28.44	0.630
		787	....	12.13	0.265	29.40	0.700
		871	....	19.79	0.419	30.67	0.772
	LL′	663	L2, M1, L3, M2.....	6.18	0.165	33.95	0.761
		789	....	11.28	0.292	34.52	0.790
		788	....	11.93	0.307	34.44	0.830
		725	....	14.76	0.375	35.10	0.824
		717	....	20.36	0.496	35.30	0.893
	MM′	667	L4, M5, M3, M4....	2.98	0.076	31.20	0.637
		790	....	9.35	0.230	32.27	0.735
		786	....	9.55	0.235	32.25	0.741
		727	....	15.78	0.375	33.10	0.786
		715	....	21.84	0.523	33.70	0.813

TABLE 1. TEST DATA FOR MOISTURE-DENSITY AND MOISTURE-CONDUCTIVITY CORRECTION CURVES—*Continued*

Species	Panels	Test No.	Composed of Boards	Moisture Per Cent by Weight Dry Basis	Moisture Lb/Bd Ft as Tested	Density Lb/Cu Ft as Tested	Conductivity "K" 75 Deg M. T.
White Oak	BB′	761	D2, D3, D3, D1	5.13	0.200	49.18	1.197
		483	....	7.18	0.276	49.54	1.221
		878	....	8.67	0.328	49.32	1.235
		892	....	10.61	0.398	49.75	1.255
	FF′	591	I1, I1, I2, I2	6.46	0.201	39.85	1.084
		486	....	7.38	0.223	39.96	1.086
		888	....	9.35	0.285	39.94	1.108
		896	....	11.43	0.343	40.16	1.122
Red Oak	BB′	939	B2, B2, B3, B3	4.83	0.184	47.85	1.180
		576	....	5.54	0.211	48.26	1.202
		492	....	6.07	0.232	48.64	1.226
		913	....	7.69	0.289	48.57	1.221
		909	....	8.76	0.326	48.47	1.231
		925	....	10.09	0.370	48.54	1.241
	CC′	940	D3, D2, D3, G1	5.46	0.187	43.35	1.120
		903	....	5.54	0.193	43.95	1.137
		577	....	5.98	0.206	43.90	1.133
		495	....	6.75	0.232	44.01	1.140
		911	....	9.13	0.307	43.91	1.156
		921	....	11.93	0.393	44.23	1.172
Yellow Birch	DD′	861	G1, G2, G3, H1	5.69	0.177	39.46	0.958
		857	....	7.75	0.238	39.71	0.973
		516	....	9.28	0.282	39.81	0.985
		901	....	15.73	0.456	40.40	1.043
	FF′	862	K1, D1, D2, D4	5.60	0.163	36.90	0.876
		890	....	9.84	0.282	37.73	0.926
		886	....	11.78	0.332	37.88	0.940
		898	....	16.90	0.461	38.38	0.977
	JJ′	662	J1, J2, J3, J4	0.66	0.023	.....	0.955
		639	....	3.28	0.111	.....	0.992
		624	....	6.43	0.215	.....	1.039
		755	....	13.24	0.432	.....	1.080
		743	....	13.64	0.447	.....	1.100
		685	....	34.83	1.026	.....	1.294
	KK′	660	K1, K2, K3, K4	0.87	0.028	39.41	.....
		637	....	3.51	0.116	40.04	0.851
		627	....	6.03	0.194	40.85	0.888
		754	....	15.48	0.468	41.98	0.960
		742	....	16.57	0.495	42.07	0.968
		705	....	27.36	0.772	43.52	.....
		683	....	30.88	0.858	44.12	.....

TABLE 1. TEST DATA FOR MOISTURE-DENSITY AND MOISTURE-CONDUCTIVITY CORRECTION CURVES—*Continued*

Species	Panels	Test No.	Composed of Boards	Moisture Per Cent by Weight Dry Basis	Moisture Lb/Bd Ft as Tested	Density Lb/Cu Ft as Tested	Conductivity "K" 75 Deg M. T.
Yellow Birch...	LL′	659	L1, L2, L3, L4......	1.69	0.063	44.98	0.960
		636	....	4.09	0.149	45.28	1.025
		625	....	6.63	0.238	46.00	1.034
		778	....	12.02	0.408	45.60	1.154
		777	....	13.82	0.465	46.01	1.154
		744	....	15.14	0.505	46.13	1.180
		704	....	22.81	0.731	47.38	1.224
		684	....	25.27	0.793	47.37	1.197
	MM′	661	M1, M2, M3, M4....	0.60	0.023	45.03	.....
		638	....	3.30	0.121	45.40	.....
		745	....	13.20	0.452	46.51	.....
		686	....	27.19	0.850	48.12	.....

The final results for density-moisture relations for the six species are shown on Fig. 7, and the final results for the conductivity-moisture relation are shown on Fig. 8.

From an examination of the density-moisture curve for the different species, it will be noted that in each case the test points lie substantially on a straight line, and, in general, the lines have the same slope for each species. There are, however, exceptions to this general rule.

Consider the case of Ponderosa Pine, Fig. 1. The lines from the data of Panels *LL′* and *XX′* have a steeper slope than the corresponding lines for Panels *HH′* and *KK′*. This indicates that, for the first two panels, the addition of moisture gives a higher rate of increase in density than for the second two panels. From this it would appear that the action of the moisture is different in the two cases, indicating that the rate of swelling or growth due to the measured moisture content in the specimen is different. An examination of the moisture-conductivity curve for Ponderosa Pine, Fig. 1, shows that the rate of increase in conductivity is greater for Panels *LL′* and *XX′* than it is for the other two panels. This general relation might be expected; yet it does not explain why there is a difference in the rate of density increase for different specimens with the same amount of moisture absorption. Considering the curve for yellow birch, Fig. 6, the relation between the conductivity-moisture curves does not definitely follow the relation between the density-moisture curves, although in this case the points on the moisture-conductivity curves are a little more scattered.

For Shortleaf Yellow Pine, the moisture-density curves, Fig. 3, are fairly uniform in slope, but the moisture-conductivity curve for Panel *JJ′* has a very much greater slope than for the other panels.

In the results obtained for West Coast Hemlock, Fig. 4, there is a close agreement between the slopes of the curves for both density-moisture and conductivity-moisture test results.

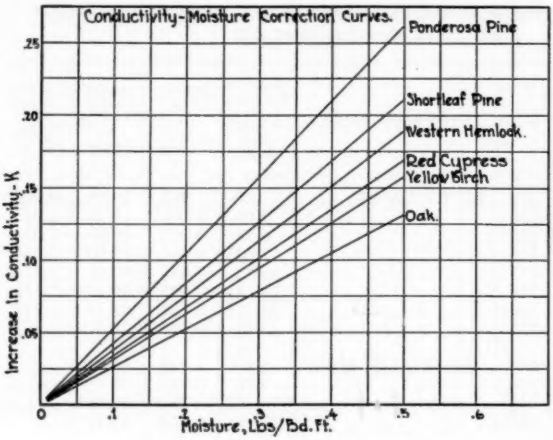


FIG. 8. THE RELATION BETWEEN CONDUCTIVITY AND MOISTURE CONTENT FOR THE REPRESENTATIVE SPECIES AS INDICATED

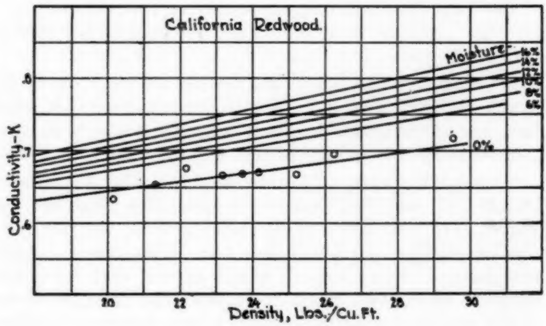


FIG. 9. CALIFORNIA REDWOOD — DENSITY-CONDUCTIVITY RELATION FOR DIFFERENT PERCENTAGES OF MOISTURE

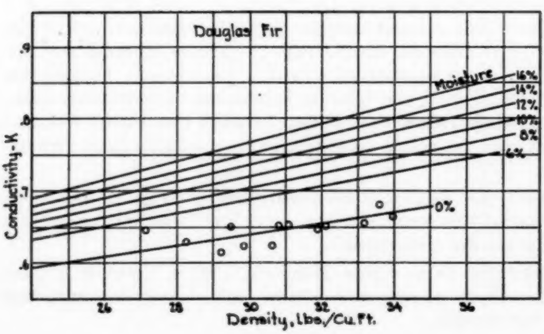


FIG. 10. DOUGLAS FIR —DENSITY - CONDUCTIVITY RELATION FOR DIFFERENT PERCENTAGES OF MOISTURE

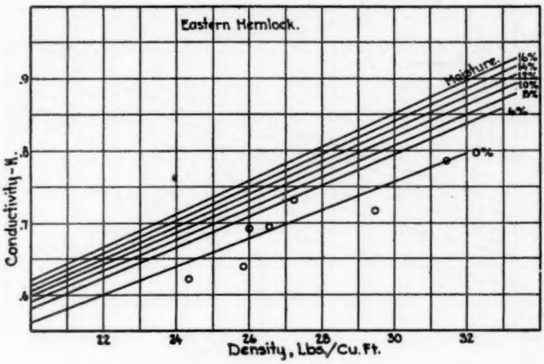


FIG. 11. EASTERN HEMLOCK — DENSITY-CONDUCTIVITY RELATION FOR DIFFERENT PERCENTAGES OF MOISTURE

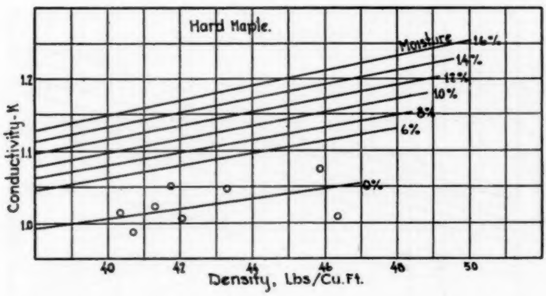


FIG. 12. HARD MAPLE —DENSITY - CONDUCTIVITY RELATION FOR DIFFERENT PERCENTAGES OF MOISTURE

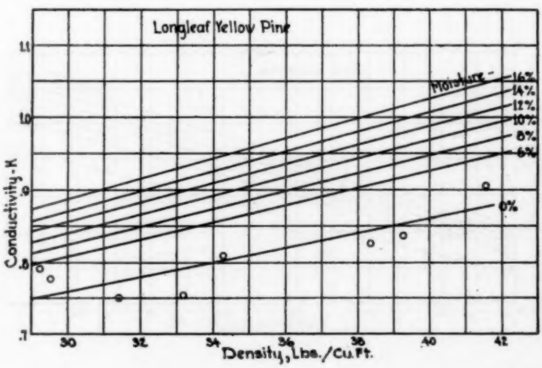


FIG. 13. LONGLEAF YELLOW PINE—DENSITY - CONDUCTIVITY RELATION FOR DIFFERENT PERCENTAGES OF MOISTURE

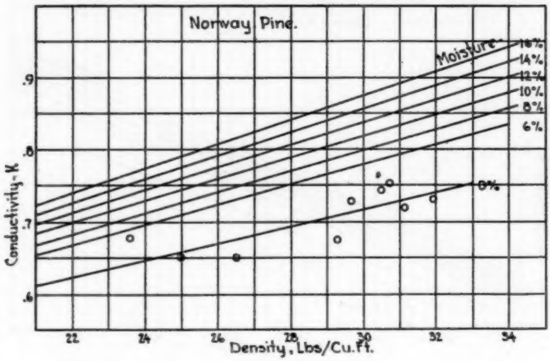


FIG. 14. NORWAY PINE—DENSITY-CONDUCTIVITY RELATION FOR DIFFERENT PERCENTAGES OF MOISTURE

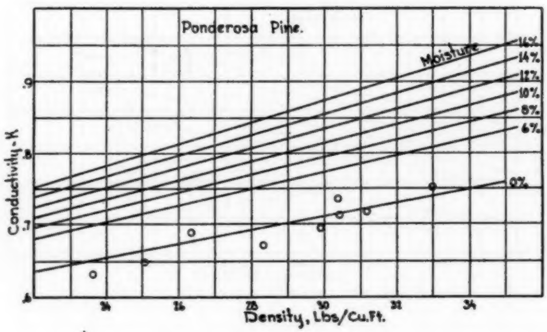


FIG. 15. PONDEROSA PINE—DENSITY-CONDUCTIVITY RELATION FOR DIFFERENT PERCENTAGES OF MOISTURE

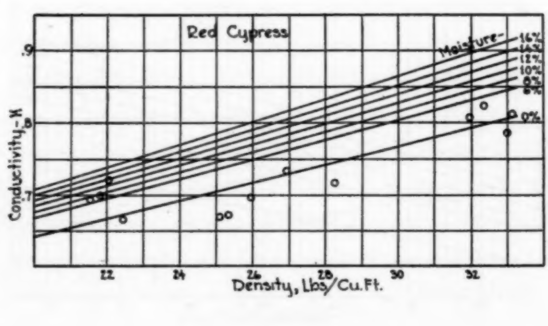


FIG. 16. RED CYPRESS—DENSITY-CONDUCTIVITY RELATION FOR DIFFERENT PERCENTAGES OF MOISTURE

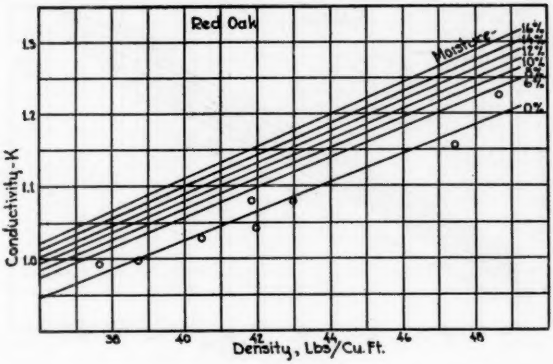


Fig. 17. Red Oak—Density - Conductivity Relation for Different Percentages of Moisture

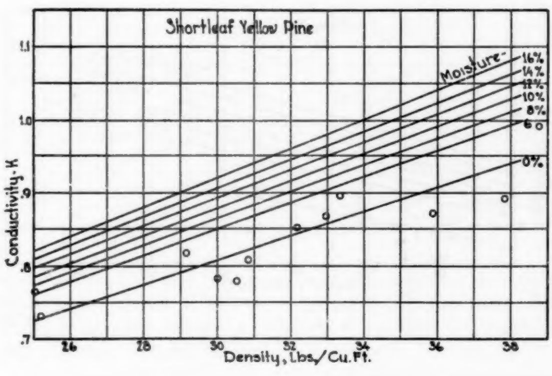


Fig. 18. Shortleaf Yellow Pine — Density - Conductivity Relation for Different Percentages of Moisture

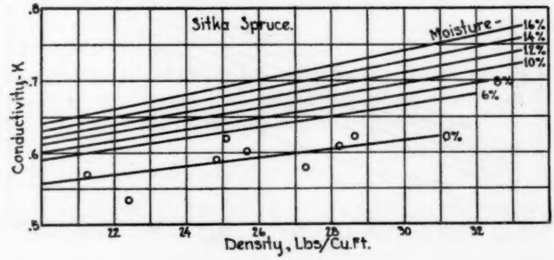


Fig. 19. Sitka Spruce —Density - Conductivity Relation for Different Percentages of Moisture

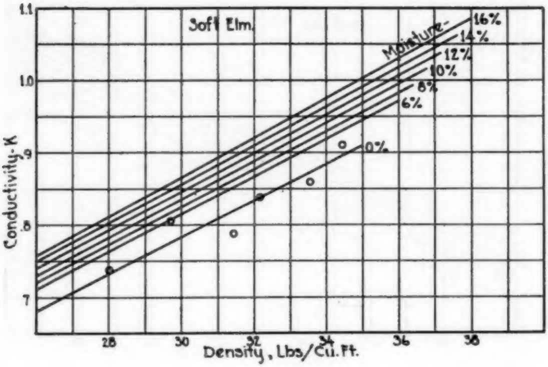
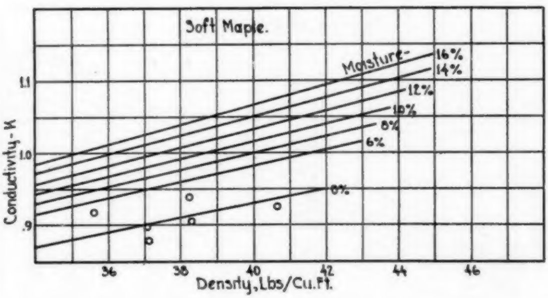


FIG. 20. SOFT ELM—DENSITY - CONDUCTIVITY RELATION FOR DIFFERENT PERCENTAGES OF MOISTURE

FIG. 21. SOFT MAPLE—DENSITY - CONDUCTIVITY RELATION FOR DIFFERENT PERCENTAGES OF MOISTURE



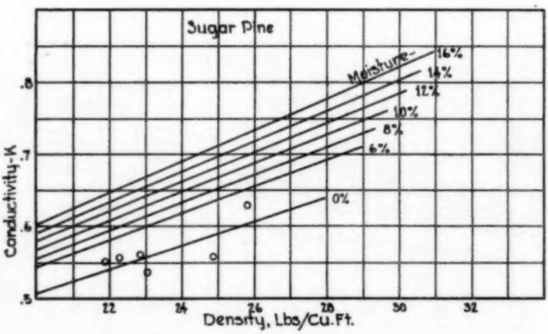


FIG. 22. SUGAR PINE—DENSITY - CONDUCTIVITY RELATION FOR DIFFERENT PERCENTAGES OF MOISTURE

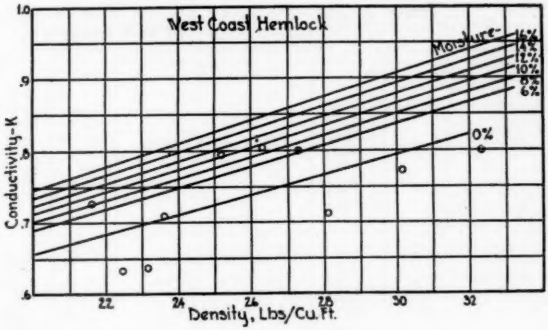


FIG. 23. WEST COAST HEMLOCK — DENSITY-CONDUCTIVITY RELATION FOR DIFFERENT PERCENTAGES OF MOISTURE

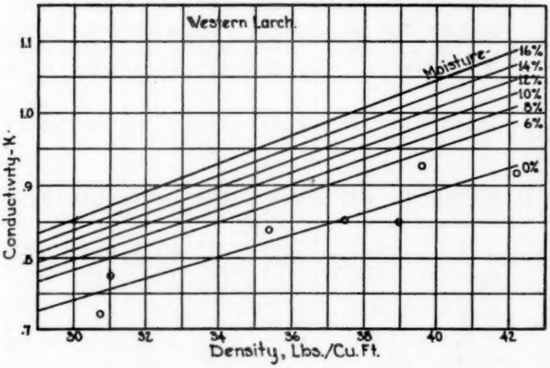


FIG. 24. WESTERN LARCH — DENSITY-CONDUCTIVITY RELATION FOR DIFFERENT PERCENTAGES OF MOISTURE

In general, it may be stated that the increase in density for the absorption of moisture is not at the same ratio for all specimens of the same species, and the conductivity-moisture relation is similar to the density-moisture relation for different samples of the same species.

By comparing the curves of Figs. 7 and 8, it will be observed that, in general, those species which have the greatest increase in density for a given increase in moisture also have the greatest increase in conductivity for a corresponding increase in moisture. Likewise, those species which are of lower density have the greatest rate of increase in density and conductivity when a given amount of moisture per board foot is added.

The density-moisture and conductivity-moisture curves as obtained for each representative species were used to convert the test results from other species of the group to results at different moisture contents as explained later.

CONDUCTIVITY-DENSITY RELATION

The conductivity-density relations as determined by the tests for the various species are shown in the curve sheets, Figs. 9 to 30, inclusive, and in the tabulated data of Table 2. For each species, the conductivities were obtained

TABLE 2. TEST DATA FOR DENSITY-CONDUCTIVITY CURVES
(Conductivities at 75 F Mean Temperature)

Species	Panels	Test No.	Composed of Boards	Density Lb/Cu Ft as Tested	Per Cent Moisture by Weight Dry Basis	Conductivity		
						As Tested	Corrected to 0 Per Cent Moisture	Corrected to 12 Per Cent Moisture
California Redwood	AA′	556	L5, L6, L3, L11....	20.67	4.92	0.661	0.634	0.700
	BB′	{395, 552}	L12, L2, L10, L8...	22.00	6.30	0.690	0.653	0.723
	CC′	551A	L7, L9, M6, M3....	22.80	5.48	0.709	0.676	0.749
	DD′	{426, 581}	M5, M12, M1, M4..	23.91	6.09	0.705	0.666	0.743
	EE′	427	M11, L4, M2, M7...	24.51	7.06	0.713	0.667	0.746
	FF′	428	H2, H3, H12, H6...	24.95	6.63	0.713	0.670	0.747
	HH′	431	H7, H1, M9, L1....	26.06	6.67	0.713	0.667	0.750
	JJ′	{433, 537, 539, 542, 544}	M8, M10, H4, H5...	27.05	5.75	0.738	0.695	0.784
	KK′	{432, 557, 15}	H8, H10, H9, H11..	30.45	6.50	0.768	0.716	0.813
Douglas Fir.....	*AB	58	A, B..............	31.67	10.75	0.733	0.625	0.746
	*AC	59	A, C..............	31.31	10.75	0.758	0.651	0.771
	*BC	60	B, C..............	35.22	10.75	0.777	0.657	0.791
	*DE	61	D, E..............	32.91	10.14	0.761	0.654	0.780
	*DF	62	D, F..............	30.98	10.52	0.720	0.616	0.734
	*EF	63	E, F..............	34.01	10.28	0.764	0.653	0.783
	*GH	64	G, H..............	36.05	10.87	0.790	0.666	0.803
	*GI	65	G, I..............	33.86	10.87	0.764	0.648	0.776
	*HI	66	H, I..............	32.56	10.87	0.738	0.625	0.749
	AA′	917	D1, D2, D3, I4.....	29.19	5.62	0.685	0.631	0.747
	BB′	918	I1, F2, F3, F4......	31.89	5.93	0.716	0.653	0.780
	DD′	920	B4, E3, H3, H4.....	34.68	5.56	0.746	0.682	0.820
	LL	598	A4, A3, A1, D2, I4..	28.08	6.53	0.707	0.647	0.756
Eastern Hemlock	AA′	632	B3, B2, B4, C1.....	32.47	7.06	0.851	0.784	0.899
	BB′	629	C2, C4, C3, B1.....	30.49	7.17	0.780	0.716	0.824
	CC′	811	A3, A4, D1, E4.....	28.22	7.63	0.754	0.731	0.831
	DD′	628	D2, A1, A2, D3.....	27.44	6.76	0.749	0.694	0.791
	EE′	631	D4, F1, E3, F2.....	26.92	7.18	0.748	0.691	0.786
	FF′	633	F4, J1, F3, I2......	26.72	6.92	0.693	0.638	0.733
	GG′	820	I3, H3, E2, I4......	25.41	7.37	0.696	0.621	0.710
	HH′	814	G1, G2, H1, H2....	23.47	6.89	0.706	0.665	0.748
Hard Maple...	AA′	646	A2, D3, D2, A1.....	42.03	5.63	1.082	1.023	1.148
	BB′	855	B1, B2, B3, B4.....	47.17	6.08	1.110	1.010	1.150
	CC′	645	C1, C2, C3, C4.....	46.72	6.10	1.146	1.075	1.214
	EE′	650	E1, E2, E3, E4.....	44.14	6.11	1.115	1.048	1.179
	FF′	859	F1, F2, F3, F4......	42.44	6.03	1.108	1.051	1.177
	GG′	652	G1, G2, G3, G4.....	41.16	6.57	1.104	1.016	1.137
	HH′	649	H1, H2, H3, H4....	41.45	5.87	1.048	0.987	1.111
	II′	858	I1, I2, I3, I4.......	42.89	5.99	1.077	1.007	1.135

* Panels tested on the 24-in. hot plate.

TABLE 2. TEST DATA FOR DENSITY-CONDUCTIVITY CURVES—*Continued*
(Conductivities at 75 F Mean Temperature)

Species	Panels	Test No.	Composed of Boards	Density Lb/Cu Ft as Tested	Per Cent Moisture by Weight Dry Basis	Conductivity As Tested	Conductivity Corrected to 0 Per Cent Moisture	Conductivity Corrected to 12 Per Cent Moisture
Longleaf Yellow Pine....	AA′	508, 354, 13, 568	B4, B3, C4, C3.....	43.11	6.58	0.999	0.906	1.076
	CC′	507	E1, E2............	41.03	7.66	0.939	0.836	0.993
	DD′	559, 511	C1, E3............	39.96	7.14	0.920	0.826	0.983
	EE′	509, 558, 569	D3, D4............	35.60	6.57	0.887	0.810	0.951
	FF′	510, 555	F2, F3............	34.47	6.65	0.829	0.754	0.889
	GG′	506, 553	I3, I4.............	32.59	6.31	0.817	0.749	0.879
	HH′	504	G2, G3............	30.78	7.39	0.850	0.776	0.896
	II′	505	H2, G1............	30.49	7.47	0.865	0.791	0.919
Norway Pine....	AA′	608	B5, B1, B2, B3.....	33.18	6.75	0.823	0.732	0.894
	BB′	821	E1, E2, E3, E4.....	32.30	6.35	0.804	0.720	0.878
	CC′	795	A3, C2, C3, C5.....	32.00	7.05	0.845	0.753	0.910
	DD′	604	A1, A2, D5, F3.....	31.56	5.87	0.820	0.743	0.899
	EE′	607	A5, C1, D4, D1.....	30.81	6.49	0.811	0.729	0.880
	FF′	609	D2, D3, F1, F2.....	30.38	6.20	0.757	0.677	0.827
	GG′	601	H1, I5, H2, H3.....	27.53	6.39	0.722	0.650	0.786
	HH′	602	H4, I2, I3, H5.....	25.93	6.38	0.720	0.652	0.780
	II′	604	G2, G3, G4, G5.....	24.44	6.08	0.739	0.678	0.799
Ponderosa Pine....	AA′	526	B4, B5, A3, A4.....	34.67	8.83	0.876	0.753	0.920
	BB′	548A	B3, A2, E1, E2.....	32.79	8.90	0.835	0.719	0.876
	CC′	807	A1, E3, B2, E4.....	31.35	8.19	0.800	0.696	0.848
	DD′	823	B1, C5, C4, C3......	29.40	6.32	0.749	0.672	0.813
	EE′	808	C2, C1, H1, D1.....	27.59	8.05	0.779	0.689	0.823
	FF′	525	F1, G3, G4, I1......	26.35	8.63	0.740	0.649	0.776
	GG′	546	D2, D3, D4, D5....	26.35	9.11	0.744	0.648	0.774
	HH′	545	H3, H4, H5, I3.....	24.83	8.73	0.718	0.631	0.751
	JJ′	391	J1, J2, J3, J4.......	31.52	6.02	0.792	0.713	0.870
	KK′	392	K1, K2, K3, K4, K5	31.42	5.68	0.810	0.751	0.876
	LL′	387	L1, L2, L3, L4......	26.54	10.75	0.815	0.725	0.825
	XX′	430	J1, J3, K1, K2, K3, K4, L1, L2, L3, L4	29.68	9.88	0.827	0.734	0.847
Red Cypress..	*AB	90	A1, A2, A3, A4, A5, B1, B2, B3, B4, B5	26.91	7.35	0.749	0.697	0.782
	*AC	98	A1, A5, C1, C5.....	26.20	7.08	0.721	0.672	0.755
	*BC	94	B1, B5, C1, C5.....	27.81	6.79	0.785	0.735	0.823
	*DE	95	D1, D5, E1, E5.....	33.01	6.97	0.870	0.809	0.913
	*DF	97	D1, D5, F1, F5.....	34.07	6.57	0.847	0.788	0.896
	*EF	96	E1, E5, F1, F5.....	34.29	7.27	0.878	0.813	0.921

* Panels tested on the 24-in. hot plate.

TABLE 2. TEST DATA FOR DENSITY-CONDUCTIVITY CURVES—*Continued*
(Conductivities at 75 F Mean Temperature)

Species	Panels	Test No.	Composed of Boards	Density Lb/Cu Ft as Tested	Per Cent Moisture by Weight Dry Basis	Conductivity		
						As Tested	Corrected to 0 Per Cent Moisture	Corrected to 12 Per Cent Moisture
Red Cypress..	*GH	99	G1, G5, H1, H5.....	22.27	6.95	0.734	0.693	0.763
	*GI	100	G1, G5, I1, I5......	22.79	6.84	0.761	0.720	0.792
	*HI	101	H1, H5, I1, I5......	22.54	6.98	0.742	0.700	0.772
	AA′	923	G1, I2, I4, I5, H4...	23.01	5.28	0.697	0.665	0.739
	BB′	924	A2, A3, A4, C2, C3..	25.79	5.58	0.709	0.670	0.753
	DD′	927	B1, B3, B4, B5, E2..	28.88	4.60	0.753	0.717	0.811
	EE′	928	D1, D2, D2, D3, E2.	33.05	4.38	0.865	0.826	0.934
Red Oak...	AA′	549A	C1, C1, C2, C2.....	49.60	5.43	1.280	1.224	1.347
	BB′	492, 576	B2, B2, B3, B3.....	48.45	5.81	1.214	1.156	1.277
	CC′	495, 577	D3, D2, D3, G1....	43.96	6.37	1.137	1.079	1.188
	DD′	854	G2, G2, G3, G3.....	42.89	7.07	1.142	1.080	1.185
	EE′	496	D1, D2, D1, G1....	42.82	6.71	1.093	1.042	1.140
	FF′	494, 575	A2, A2, A3, A3....	41.28	5.58	1.076	1.028	1.131
	GG′	550A	A1, I2, I3, I3.......	39.51	5.76	1.045	0.998	1.096
	HH′	498, 527, 529, 531	H1, H3, I1, I2......	38.43	5.87	1.040	0.993	1.089
Shortleaf Yellow Pine....	AA′	436, 565	A3, C1............	40.35	7.24	1.086	0.990	1.148
	BB′	437, 564	A2, B3............	39.18	6.13	0.971	0.892	1.047
	CC′	441	B2, C2............	37.46	7.72	0.964	0.870	1.018
	DD′	434, 14	D3, D1............	34.16	6.33	0.937	0.866	1.001
	EE′	442	E2, E1............	30.30	6.72	0.883	0.816	0.936
	FF′	435, 567	F2, F3............	34.51	5.88	0.960	0.893	1.031
	GG′	445, 570	G2, G3............	25.64	5.90	0.795	0.745	0.847
	HH′	562	H1, H3............	25.72	4.55	0.803	0.764	0.867
	II′	455, 572	I1, I2.............	26.12	6.04	0.786	0.732	0.839
	JJ′	757	J2, J3, K3, L3......	31.76	6.85	0.851	0.779	0.905
	KK′	390	J1, K1, K2, K3.....	33.42	6.64	0.923	0.851	0.983
	LL′	804	L1, L1, L2, J2......	31.21	6.83	0.854	0.784	0.907
	XX′	429	J1, J3, K1, K2, L2, L3.............	32.03	6.75	0.877	0.807	0.932
Sitka Spruce..	BB′	477, 579	A1, C1, C2, C3.....	29.50	6.26	0.677	0.623	0.727
	CC′	475, 580	A4, B1, B2, B3.....	28.10	6.26	0.630	0.578	0.678

* Panels tested on the 24-in. hot plate.

TABLE 2. TEST DATA FOR DENSITY-CONDUCTIVITY CURVES—*Continued*
(Conductivities at 75 F Mean Temperature)

Species	Panels	Test No.	Composed of Boards	Density Lb/Cu Ft as Tested	Per Cent Moisture by Weight Dry Basis	Conductivity		
						As Tested	Corrected to 0 Per Cent Moisture	Corrected to 12 Per Cent Moisture
Sitka Spruce...	DD′	935	A2, A3, B4.........	29.09	6.36	0.664	0.609	0.713
	FF′	937	G4, H1, H2, I1, I2, I3	26.45	6.55	0.652	0.601	0.695
	GG′	473, 11, 12	F4, G1, G2, G3.....	25.78	6.01	0.665	0.619	0.711
	HH′	938	H2, I1, I2, I3.......	25.57	6.35	0.638	0.590	0.681
	JJ′	467, 19	J1, J2, J3, J4.......	23.10	6.58	0.579	0.534	0.616
	KK′	474, 20	K1, K2, K3, K4....	21.93	6.93	0.614	0.569	0.647
Soft Elm..	AA′	523	E4, D2, D3, F3.....	35.26	7.67	0.976	0.910	1.013
	BB′	797	B3, C4, D1, E3.....	34.52	7.29	0.920	0.859	0.960
	CC′	518	C3, D4, I1, A4......	32.91	7.58	0.900	0.839	0.935
	DD′	520	A1, A2, A3, C2.....	32.27	7.83	0.854	0.787	0.890
	EE′	519	H3, H4, I3.........	30.39	7.51	0.860	0.805	0.894
	FF′	521	I4, G1, G2, G3, H2.	28.61	7.18	0.787	0.737	0.821
Soft Maple...	AA′	694	A4, G4, A3, A1.....	39.03	6.03	0.962	0.904	1.020
	DD′	696	D1, D2, D3, D4....	36.33	6.54	0.976	0.917	1.024
	EE′	698	E1, E2, E3, E4.....	37.83	6.07	0.936	0.879	0.991
	FF′	653	F1, F3, B4, C1......	37.77	6.04	0.952	0.896	1.008
	GG′	697	G1, G2, G3, H4.....	38.98	6.33	0.999	0.938	1.054
	II′	695	I1, I2, I3, I4.......	41.41	6.12	0.988	0.925	1.048
Sugar Pine.	AA′	500, 586	D1, D2, D4, F2.....	27.09	8.44	0.721	0.629	0.760
	BB′	815	F3, G3, G2, G4.....	26.12	8.43	0.648	0.559	0.686
	CC′	499, 22	E1, H2, H3, H4....	24.22	8.60	0.621	0.537	0.654
	DD′	502, 563	E2, E4, I3, E3.....	23.96	8.27	0.642	0.562	0.678
	EE′	503, 583	B1, B2, B3, B4.....	23.38	8.34	0.636	0.557	0.670
	FF′	501	A3, C1, C2, C3.....	23.08	8.86	0.634	0.552	0.663
West Coast Hemlock.	AA′	465, 574	A2, A3, A4, B1.....	33.18	5.58	0.755	0.700	0.819
	BB′	462	A5, B2, B4, B5.....	31.07	6.42	0.732	0.673	0.784
	FF′	463, 538, 560	B3, F1, F2, F3.....	28.98	6.32	0.667	0.613	0.716
	GG′	460, 540, 10	G1, G2, G3, G4.....	27.06	5.84	0.752	0.704	0.803
	HH′	476, 561	H2, H3, H4, H5....	23.90	6.81	0.585	0.537	0.6[illegible]2

TABLE 2. TEST DATA FOR DENSITY-CONDUCTIVITY CURVES—*Continued*
(Conductivities at 75 F Mean Temperature)

Species	Panels	Test No.	Composed of Boards	Density Lb/Cu Ft as Tested	Per Cent Moisture by Weight Dry Basis	Conductivity As Tested	Conductivity Corrected to 0 Per Cent Moisture	Conductivity Corrected to 12 Per Cent Moisture
West Coast Hemlock	II′	{461, 17}	I2, I3, I4, I5	23.14	6.19	0.576	0.533	0.616
	JJ′	666	J2, K5, L4, L2	24.36	5.38	0.657	0.610	0.716
	KK′	664	K5, K4, J3, K3	22.32	5.30	0.669	0.627	0.723
	LL′	663	L2, M1, L3, M2	28.28	6.18	0.761	0.699	0.820
	MM′	665	L4, M5, M3, M4	26.53	9.04	0.779	0.694	0.806
Western Larch	AA′	458	A3, C3, C4, F3	44.62	10.00	1.059	0.917	1.088
	BB′	454	A2, C5, D5, F4	41.86	9.66	1.059	0.926	1.092
	CC′	449	A4, B2, C2, D4	41.28	10.46	0.985	0.848	1.006
	DD′	450	B4, D3, G2, H5	39.68	10.30	0.986	0.857	1.008
	KE′	448	D2, F5, G3, H2	37.50	10.30	0.962	0.839	0.982
	FF′	451	I1, I2, I3, C5	32.95	10.81	0.889	0.775	0.901
	GG′	459	E1, E2, E3, E4	32.81	10.61	0.831	0.721	0.845
Western Red Cedar	*AB	67	A, B	26.40	11.00	0.774	0.700	0.781
	*AC	68	A, C	25.09	11.58	0.700	0.627	0.703
	*DE	70	D, E	22.50	10.15	0.617	0.558	0.627
	*DF	69	D, F	22.62	9.97	0.616	0.558	0.628
	*EF	71	E, F	22.80	9.70	0.612	0.555	0.625
	*GH	79	G, H	21.82	8.25	0.603	0.556	0.624
	*GI	80	G, I	21.86	8.14	0.560	0.514	0.582
	*HI	78	H, I	22.31	8.46	0.604	0.555	0.625
	HH′	951	A4, A2, A3, A1, A3	28.29	5.05	0.817	0.779	0.870
	LL′	595	G4, G2, G2, G4, G3	20.70	6.43	0.585	0.550	0.616
White Ash	AA′	592	A1, B1, B2, F1	45.98	8.59	1.163	1.084	1.195
	BB′	470	A1, A2, A2, A3	45.42	9.73	1.123	1.035	1.144
	DD′	{482, 593}	B4, E1, E1, E3	42.40	8.81	1.019	0.944	1.046
	EE′	{484, 594}	B3, E6, F3, I1	41.82	8.46	1.055	0.984	1.085
	FF′	478	I2, H2, H3, E5	38.67	8.66	0.939	0.872	0.965
	GG′	480	E5, I2, I3, H1	38.20	8.56	1.021	0.955	1.047
	HH′	481	E2, E2, G2, G2	32.14	8.13	0.908	0.855	0.933
White Fir	*AB	49	A, B	21.92	8.80	0.565	0.509	0.585
	*AC	50	A, C	22.37	8.28	0.573	0.519	0.597
	*BC	51	B, C	22.48	8.42	0.588	0.533	0.612
	*DE	52	D, E	23.38	8.50	0.594	0.536	0.618
	*DF	53	D, F	24.15	8.37	0.595	0.536	0.620
	*EF	54	E, F	25.32	8.49	0.626	0.564	0.652
	*GH	55	G, H	26.50	8.49	0.662	0.597	0.689
	*GI	56	G, I	25.10	8.60	0.656	0.593	0.681
	*HI	57	H, I	27.20	8.35	0.693	0.627	0.722

* Panels tested on the 24-in. hot plate.

TABLE 2. TEST DATA FOR DENSITY-CONDUCTIVITY CURVES—*Continued*
(Conductivities at 75 F Mean Temperature)

Species	Panels	Test No.	Composed of Boards	Density Lb/Cu Ft as Tested	Per Cent Moisture by Weight Dry Basis	Conductivity As Tested	Conductivity Corrected to 0 Per Cent Moisture	Conductivity Corrected to 12 Per Cent Moisture
White Oak.....	AA′	{487, 23}	E1, E1, E2, E2.....	51.35	6.75	1.244	1.173	1.300
	BB′	483	D2, D3, D3, D1....	49.54	7.18	1.221	1.148	1.269
	CC′	{489, 588}	D1, D2, F3, H3.....	48.61	7.00	1.176	1.106	1.226
	DD′	{485, 589}	F1, F2, F2, H1.....	44.03	7.26	1.180	1.114	1.222
	EE′	{488, 590}	G2, G2, G3, G3.....	42.79	6.74	1.136	1.077	1.183
	FF′	{486, 591}	I1, I1, I2, I2........	39.99	6.92	1.085	1.028	1.127
Northern White Pine....	AA′	612	A3, A2, A1, A4.....	32.40	6.25	0.880	0.797	0.957
	BB′	617	A5, C3, C4, C2.....	31.28	6.78	0.911	0.824	0.978
	CC′	616	E1, E2, E3, E4.....	27.95	6.24	0.758	0.686	0.824
	DD′	614	E5, B3, B4, D5.....	26.84	6.93	0.795	0.719	0.851
	EE′	611	B1, B2, C1, F4.....	26.79	6.65	0.794	0.721	0.853
	FF′	615	F5, B5, F2, F3.....	24.47	6.55	0.704	0.638	0.758
	GG′	618	F1, G4, G5, I2......	23.87	6.14	0.657	0.597	0.715
	HH′	622	D3, D1, G3, I1.....	23.01	6.54	0.662	0.600	0.714
	II′	613	H1, H2, H3, H4....	21.02	6.32	0.663	0.608	0.711
Yellow Birch...	AA′	515	B1, B2, B3, C2.....	45.34	9.71	1.022	0.917	1.047
	BB′	514	F1, F2, F3, F4......	43.94	9.14	1.005	0.909	1.035
	CC′	512	I1, I2, I4, H2.......	41.54	9.20	0.960	0.868	0.988
	DD′	516	G1, G2, G3, H1.....	39.81	9.28	0.985	0.896	1.011
	EE′	513	E2, E3, D3, G4.....	38.82	9.49	0.856	0.769	0.979
	FF′	890	E1, D1, D2, D4....	37.73	9.84	0.926	0.837	0.945
	KK′	627	K1, K2, K3, K4....	40.85	6.03	0.888	0.827	0.949
	LL′	625	L1, L2, L3, L4.....	46.00	6.63	1.034	0.959	1.095

for the different densities at moisture content as received which varied from 4½ per cent to 11 per cent. The per cent moisture as tested is shown in the tabulated data. The thermal conductivity and density values obtained in the tests were corrected to 0 per cent moisture and plotted on the curve sheets, Figs. 9 to 30, inclusive. The conductivity curve for 0 per cent moisture was drawn and the values from this curve were used to determine the other conductivity values ranging up to 16 per cent moisture.

In making the corrections in conductivity and density for 0 per cent moisture, the curves for moisture-density and moisture-conductivity, Figs. 7 and 8, were taken as determined for the species which was used as the representative of the particular material under test. From the density-moisture curve, the density at 0 per cent moisture was determined, and from the conductivity-moisture curve the conductivity at 0 per cent moisture was determined, thus giving the coordinates for conductivity-density corresponding to the particular sample

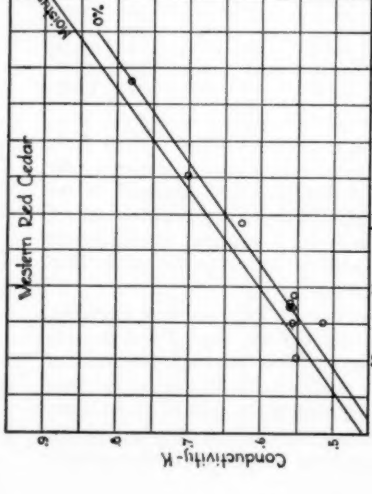


FIG. 25. WESTERN RED CEDAR

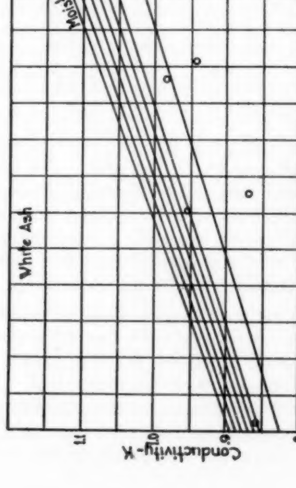


FIG. 26. WHITE ASH

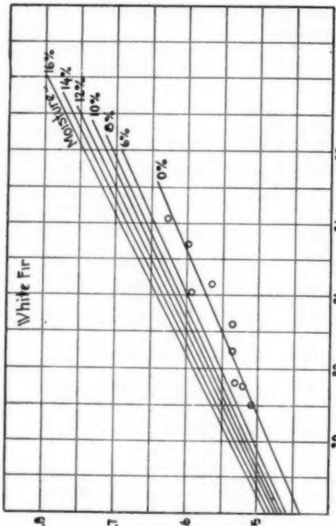


FIG. 27. WHITE FIR

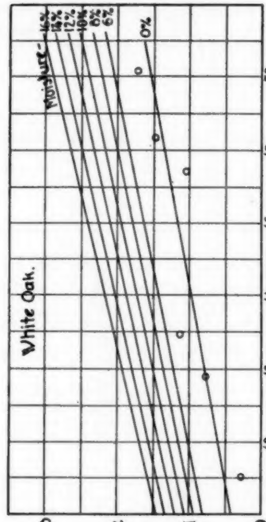


FIG. 28. WHITE OAK

CHARTS SHOWING DENSITY-CONDUCTIVITY RELATION FOR DIFFERENT PERCENTAGES OF MOISTURE

tested. The points for all tests were corrected in this manner, and the 0 per cent curve was drawn through these points by the method of least squares.

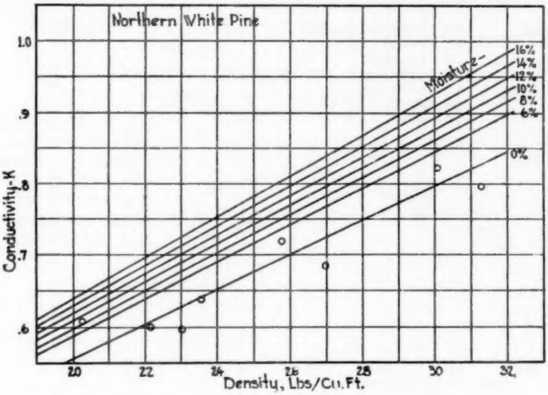
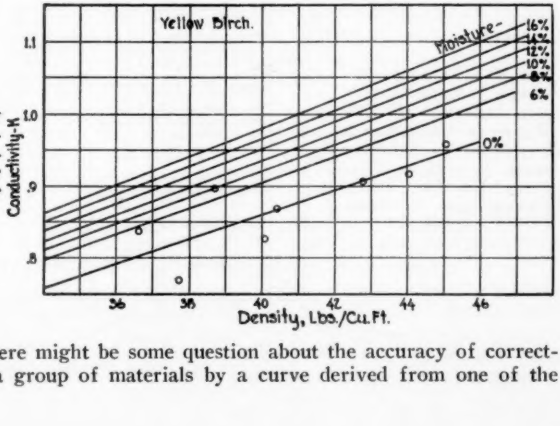


FIG. 29. NORTHERN WHITE PINE — DENSITY - CONDUCTIVITY RELATION FOR DIFFERENT PERCENTAGES OF MOISTURE

The points for the curves of higher percentages of moisture for a given species were calculated from the 0 per cent curve in the following manner: For a given density in pounds per cubic foot and a given moisture content, the weight of moisture in pounds per board foot was calculated. From this value, the increase in density in pounds per cubic foot was obtained from the representative species, Fig. 7. In like manner, the increase in conductivity for the calculated moisture content in pounds per board foot was determined from Fig. 8. These increases in density and conductivity were added to the density and conductivity values for the original selected point, and the new values plotted on the curve sheet as a new point on the conductivity-density relation for the given percentage of moisture.

Since these lines were plotted as straight lines, it is evident that the calculation of two points for each moisture content was sufficient. As has been

FIG. 30. YELLOW BIRCH — DENSITY-CONDUCTIVITY RELATION FOR DIFFERENT PERCENTAGES OF MOISTURE



previously stated, there might be some question about the accuracy of correcting the values for a group of materials by a curve derived from one of the

group as representative. It should be pointed out, however, that the tests were made at moisture contents ranging from 6 to 10 per cent. Thus, if there was

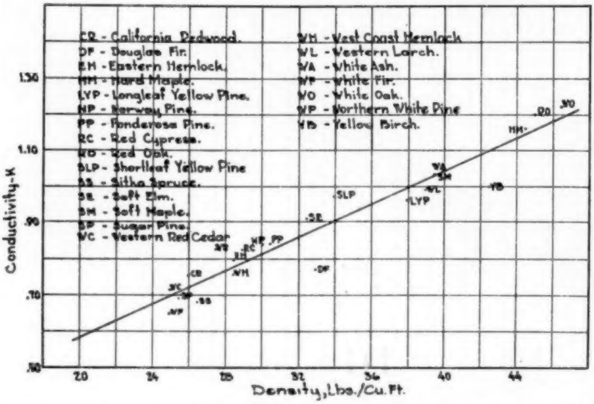


FIG. 31. RELATION BETWEEN CONDUCTIVITY AND DENSITY AT 12% MOISTURE FOR AVERAGE DENSITIES OF SPECIES AS TESTED

an error in correcting back to the 0 per cent moisture, this error would be eliminated in correcting the given point back to a percentage of moisture corresponding to test conditions.

In correlating the data of Tables 1 and 2 with the curve sheet, it should be observed that the moisture per cent by weight shown in the tables is given as the percentage by weight of the dry material in the specimen, whereas in the curves of Figs. 9 to 30, inclusive, the percentages of moisture are based upon the total weight per cubic foot and not on the dry material. Thus from the curves, the conductivity for any density and moisture content from 0 to 16 per cent may be read directly.

In order to get some comparison between the density-conductivity relation for the different species, curves on Figs. 31 and 32 were plotted. The points

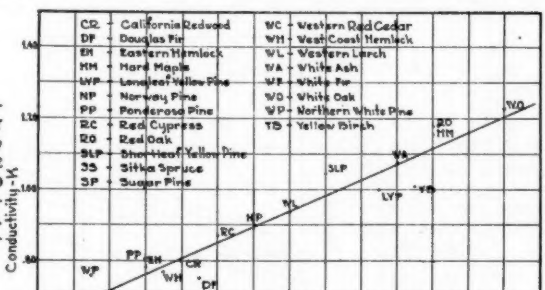


FIG. 32. RELATION BETWEEN CONDUCTIVITY AND DENSITY AT 12% MOISTURE FOR AVERAGE DENSITIES AS SELECTED FROM TECHNICAL BULLETIN No. 158, U. S. DEPARTMENT OF AGRICULTURE

for Fig. 31 represent the conductivity at 12 per cent moisture for the average density of each species as tested. For Fig. 32 the points represent the con-

ductivities as taken from the 12 per cent moisture lines, but the densities were the averages for each species as given in *Technical Bulletin* No. 158 of the U. S. Department of Agriculture. The densities used for the curves of Fig. 32 were as follows:

SPECIES	DENSITY, LB PER CU FT
California Redwood	30
Douglas Fir	31
Eastern Hemlock	28
Hard Maple	44
Longleaf Yellow Pine	41
Norway Pine	34
Ponderosa Pine	28
Red Cypress	32
Red Oak	44
Shortleaf Yellow Pine	38
Sitka Spruce	28
Sugar Pine	25
Western Red Cedar	23
West Coast Hemlock	29
Western Larch	36
White Ash	42
White Fir	26
White Oak	48
Northern White Pine	25
Yellow Birch	43

By comparing the densities of the materials tested with the average densities taken from *Bulletin* No. 158, it will be noted that, in a majority of cases, the selected values fall within the range of test values; in a few cases, such as California Redwood, Norway Pine, Shortleaf Pine, Soft Elm, White Ash, and White Fir, the densities of the materials tested were of a lower average than those given in the bulletin. The final curves of density-conductivity relation, however, for the several species as shown in Figs. 31 and 32 are substantially the same.

The conductivity values for all species fall sufficiently close to the straight line to indicate that for practical purposes the straight line relation may be used when the moisture contents are the same.

In making these tests and preparing this report, acknowledgment is due Mr. Frank P. Cartwright, Chief Engineer of the *National Lumber Manufacturers' Association,* for his suggestions throughout the work, as well as for the initial selection and preparation of the samples; to Mr. A. B. Algren for his assistance in supervising the laboratory program; and to Messrs. W. A. Eckley and L. A. Clousing, who conducted the greater part of the tests by the hot plate method.

DISCUSSION

E. R. QUEER (WRITTEN): Anyone familiar with thermal conductivity measurements by the guarded electric hot plate method will appreciate the great amount of time and patience expended in collecting the data reported in this paper.

The guarded electric hot plate offers the best known method of obtaining thermal conductivities of building and insulating materials in a dry condition. Reliable data can also be obtained at low moisture contents. There is, however, some question as to the interpretation of the results of this method when the materials contain quantities of moisture. The desirability of making these measurements quite rapidly

was pointed out in a previous paper.[1] In using the guarded electric hot plate, as much as 8 to 10 hours may be necessary to obtain thermal equilibrium and to maintain it for 4 or 5 hours. During the first few hours the moisture and temperature conditions in the specimen are in a transient state. As soon as heat is applied on one side and removed from the other the moisture moves in the direction of the heat flow. For a given temperature gradient a corresponding moisture gradient will exist throughout a specimen: a high concentration existing in the low temperature fibers and a low concentration in the high temperature fibers. Although some of the specimens in these tests showed a slight loss of moisture during the conductivity determination, there is no indication of how the moisture was distributed within the sample while a test was being made. After heat at a given temperature has been applied to a specimen for some time, the moisture distribution becomes stabilized and thermal equilibrium can be established for a test. Under these conditions the conductivity of the dry fibers will be lower than that of the more moist or cold fibers. A group of properly placed thermocouples perpendicular to the hot plate surface, in the case of a moist specimen probably will not show a straight line temperature gradient through the specimen with the hot and cold side temperatures in equilibrium.

From a practical standpoint the tests are of value since they represent some conditions that actually occur in service. The conductivities reported in this paper represent an average conductivity for a specimen with a given high and low side temperature and a given moisture content. Although these tests were made at a 75 F mean temperature, it may be possible that changing the high and low side temperatures from test to test to maintain the same mean temperature will cause a changeable moisture gradient and alter the results.

F. B. ROWLEY: It should be noted that the effect of moisture on conductivity was carefully studied for several species of wood. The results of these studies are shown in Figs. 1 to 6 inclusive, and in Fig. 8. From these, and particularly from Fig. 8, it will be noted that there is a straight line relation between moisture content and conductivity. Thus, even though there should be some slight movement of moisture in the specimen during the test it would serve to increase the conductivity on the side to which the moisture was moving at the same rate as it decreased it on the side from which it was moving, giving the same result for overall conductivity as long as the moisture remained in the specimen. As stated in the paper, there was practically no loss of moisture from the test specimens throughout the tests. As Professor Queer stated, the tests were made at 75 F mean temperature and it was necessary to make some variations in the high and low temperatures to maintain the 75 F mean temperature. There is, however, no logical reason why this adjustment would alter the results. Many tests have been made which prove that for a given mean temperature the conductivity will not be changed by raising or lowering the differences between the extremes of high and low temperatures.

E. C. RACK: I feel that the work on this subject is of much value. In view of the fact that so little data are available, it is hoped that similar studies may be continued in order to more definitely extend our sources of available information on this subject.

As a matter of information and record, it is requested that Professor Rowley advise as to the thickness of specimens, as well as the approximate surface temperatures maintained during the progress of tests.

PROFESSOR ROWLEY: A mean temperature of 75 F was maintained with an average 100 F high temperature and 50 F low temperature. The average thickness of the specimen was 1 in. It was necessary in some cases, however, to plane the test specimens slightly less than 1 in. in thickness in order to get uniform surfaces for contact with the hot and cold plates.

[1] Effect of Moisture on the Heat Transmission in Insulating Materials, by L. F. Miller, *Refrigerating Engineering*, Nov. 1927.

No. 965

PHYSIOLOGIC CHANGES DURING EXPOSURE TO IONIZED AIR

By C. P. YAGLOU † (*MEMBER*), A. D. BRANDT ‡ AND L. C. BENJAMIN § (*NON-MEMBERS*) BOSTON, MASS.

This paper is the result of research sponsored by the AMERICA SOCIETY OF HEATING AND VENTILATING ENGINEERS and conducted at Harvard University, School of Public Health, Boston, Mass.

THIS is the third paper of a series on the problem of ionization in relation to ventilation and health.[1, 2] It deals with the physiologic effects of artificially ionized air on human subjects at rest under basal and normal dietary conditions.

The first comprehensive study of the biologic and clinical effects of ionized air was published in 1931 by Dessauer and his associates.[3, 4] The ions used in this previous work consisted of charged submicroscopic particles of magnesium oxide dust having mobilities between 0.007 and 0.0018 cm/sec/volt/cm. Air with an ion count of 10^6 per cubic centimeter was administered to patients by means of a funnel placed over the face of the subject, who inhaled the ionized air for periods varying from 15 min to an hour.

Under these conditions Dessauer found an apparent opposite physiologic effect of positive and negative ions on both normal subjects and patients of the clinic. As a rule, positive ions increased the respiration rate, basal metabolism, and blood pressure, while negative ions decreased these functions. In most instances the inhalation of positive ions resulted in headaches, dizziness, nausea, and a feeling of fatigue; negative ions, on the other hand, produced a feeling of exhilaration and an apparent improvement in health.

The study to be described in the present paper began about the middle of 1930. Preliminary experiments with small positive ions yielded results somewhat in accord with Dessauer's; however, the whole series was discarded

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‡ Research Fellow.
§ Technician.

[1] Changes in Ionic Content of Air in Occupied Rooms Ventilated by Natural and Mechanical Methods, by C. P. Yaglou, L. C. Benjamin and S. P. Choate, A. S. H. V. E. TRANSACTIONS, Vol. 38, 1932, p. 191.

[2] The Influence of Respiration and Transpiration on Ionic Content of Air of Occupied Rooms, by C. P. Yaglou, L. C. Benjamin, and A. D. Brandt, *Journal Industrial Hygiene*, 1933, *15*, 8.

[3] Zehn Jahre Forschung auf dem Physikalisch-Medizinischen Grenzgebiet, by F. Dessauer, Leipzig, Georg Thieme, 1931.

[4] Life and Our Atmosphere (Abridged translation of Dessauer's book), by F. E. Hartman, *Aerologist*, 1931, *7*, Nos. 10, 11, 12, and 1932, *8*, Nos. 1 and 2.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Hotel Statler, Detroit, Mich., June, 1933, by C. P. Yaglou.

because of a suspected vitiation by ozone, which was present in appreciable concentrations as a by-product of early ionization methods. Subsequent experiments (since January, 1931) with improved apparatus have not entirely confirmed the preliminary observations and Dessauer's conclusions.

Character of Ions Used and Method of Administration

The ions used in the present study were of the small molecular type having on an average a mobility of 1.3 cm/sec/volt/cm in the breathing zone of the experimental chamber. The entire quantity of air supplied to the room was ionized so that the subjects were wholly exposed to the influence of the ions. The concentration of positive and negative ions in different experiments was varied from a minimum of about 5,000 per cubic centimeter of air to a maximum of 1,500,000. (Normal air contains from 50 to 800 small ions per cc.) Respectively within the limits of these two concentrations from 50 to 99 per cent of the ions were absorbed by the rubber diaphragm of the respiration valves during the period of determining the subjects' oxygen consumption. In other words, during this period (10 to 20 min) the subject breathed air containing only a small fraction of the ions present in the room air. Concentrations in excess of 1,000,000 unipolar ions per cubic centimeter of air proved impracticable owing to the accumulation of bothersome static charges on the bodies of subjects, on apparatus, etc. All ion counts were made in the breathing zone of the patients, using a suitable counter similar to that described in the first paper.[1]

The ionizers employed were capable of producing either positive or negative ions alone, to the exclusion of the other, or mixtures of positive and negative ions in any desired quantities and proportions. In the experiments with unipolar ions the concentration of the unwanted ions was much less than that present in ordinary air, owing to rapid neutralization by the artificial ions.

Method of Study

All experiments were carried out in the psychrometric chamber of the school, which is equipped with a complete air conditioning apparatus. Outdoor air, heated to a comfortable temperature, according to the requirements of the subjects, was circulated through the room at a rate as high as 35 air changes per hour (1,800 cfm) in order to maintain normal chemical composition in the inspired air. No attempt was made to humidify the air in cold weather nor to cool it in warm weather; the purpose of this was to avoid as much as possible the complicating influence of adaptation to temperature and humidity changes.

The experiments were divided into three series, (*a*) those under standard basal conditions, (*b*) those from two to four hours after breakfast, and (*c*) those three to five hours after a light lunch. Most of the observations were carried out in the morning between 9 and 12:30. The general plan was as follows: the subjects, one to three at a time, came to the laboratory about 9 a. m. (or 2 p. m.) and lay quietly and comfortably on cots for a preliminary period of one to two hours. During this resting period, the pulse rate, blood pressure, and respiration rate were taken at intervals until they assumed a steady level. At the end of this time the mouth temperature and metabolism

were determined at least once. In many experiments arterial and finger blood was taken for a study of blood changes after completing all the foregoing observations. These data will be presented in another paper.

Having established a definite base line in this way, the ionizer was turned on, and adjusted so as to produce the desired sign and concentration of ions. All other conditions were kept unaltered. As a rule, exposure to the ionized air was limited to about an hour, though in a few experiments it varied from half an hour to three hours. During this ionization period all the observations listed were repeated at intervals, and near the end of the test the subjects were questioned about their sensations and general impressions.

In some of the experiments showing appreciable physiologic changes on exposure to ionized air, the observations were continued for one-half to one hour after the ionization period, without letting the subjects know that the ionizer was turned off. In a special series of control experiments the subjects assumed that the air was being ionized, though as a matter of fact they were exposed only to normal air throughout the test period. The chief purpose of these control experiments was to study the influence of the prolonged rest, food, the psychic factor, etc., on the physiologic functions.

With the exception of four instances, the observations under basal conditions were made according to the generally accepted standard practice following Benedict's [5] outline of prerequisites. Briefly these are (*a*) that the patient must have no food for at least 12 hours before the observations; (*b*) that he should be lying quietly and comfortably in an environment which is neither too warm nor too cool; (*c*) that he should not have a febrile temperature; (*d*) that he should be in complete muscular repose and without psychic disturbance and tension.

In the four exceptional cases the subjects had a very small breakfast three hours or more before making observations. This had little or no effect on the basal functions, as was also the case in Benedict's [6] and DuBois's [7] experiments, and it improved immensely the psychologic attitude of the subjects toward the experiment.

The metabolism was determined by the Tissot open-circuit method except in a few instances in which the closed-circuit oxygen consumption method was employed. The subject inhaled room air through a special *inhalator mask* which covered closely the mouth and nose. He breathed normally by nose or mouth, and the expired air was collected in a spirometer after two rinsings. A pair of Loven's [8] rubber diaphragm valves were used to separate the inspired from the expired air. The expired air was sampled in mercury sampling tubes and analyzed in a Haldane gas analysis apparatus on the day of the experiments.

New subjects who had no experience in breathing through a mask were given an opportunity to try it for short periods and become accustomed to the routine before making actual measurements. In a number of instances the data of the first and second determination had to be discarded, either because

[5] Basal Metabolism Data on Normal Men and Women (Series II) with Some Considerations on the Use of Prediction Standards, by F. G. Benedict, *American Journal Physiology*, 1928, *85*, 608.
[6] A Permissible Breakfast Prior to Basal Metabolism Measurements, by C. G. Benedict and F. G. Benedict, *Boston Medical and Surgical Journal*, 1923, *188*, 849.
[7] The Effect of a Small Breakfast on Heat Production, by G. F. Soderstrom, D. P. Barr and E. F. DuBois, *Arch. Intern. Med.*, 1918, *21*, 613, 619.
[8] The Effect of the Ingestion of Foodstuffs on the Respiratory Exchange in Pulmonary Tuberculosis, by W. S. McCann, *Arch. Intern. Med.*, 1921, *28*, 847.

TABLE 1. PHYSIOLOGIC CHANGES DURING EXPOSURE TO IONIZED AIR

(Summary of Individual Observations on 15 Subjects)

Subject No.	Sex	Age	Weight, Kg	Height, Cm	Date of Experiment	Ions per Cubic Centimeter of Air		Exposure to Ionized Air, Min	Dietary Condition	Calories per Hour				Respiration Rate			Pulse Rate			Blood Pressure Mm Hg						Oral Temp. F		Remarks
																				Before		During		After				
						Positive	Negative			Harris-Benedict Basal Standard	Before Ionization	During Ionization	After Ionization	Before	During	After	Before	During	After	Systolic	Diastolic	Systolic	Diastolic	Systolic	Diastolic	Before	During	
1	M	46	60.0	180	4-21-33	....	250,000	37	Basal	62	62	61	..	21	15	..	67	63	..	110	73	104	68	...	..	97.7	97.8	Apparently normal.
1	..	..	60.0	...	4- 7-33	150,000	....	45	Basal	62	61	53	..	22	19	..	69	64	..	89	60	104	72	...	..	98.3	98.2	Apparently normal.
1	..	45	61.0	...	11- 1-32	100,000	....	46	After breakfast	..	75	69	..	20	15	..	68	66	..	115	65	112	71	...	..	98.0	97.7	Apparently normal.
1	..	..	60.0	...	1-20-31	20,000	3,000	70	After breakfast	..	56	67	60	21	17	18	83	70	74	117	67	114	70	118	69	98.4	98.0	Subject in run-down condition.
2	F	27	58.5	161	3-28-33	....	100,000	65	Basal	55	44	55	..	13	12	..	65	66	..	83	54	92	57	...	..	98.4	98.3	Apparently normal.
2	..	..	59.0	...	2-28-33	500,000	....	70	Basal	55	57	57	..	10	15	..	63	63	..	82	53	92	57	...	..	98.3	98.3	
2	..	26	59.0	...	1-26-31	10,000	3,000	63	After lunch	..	70	65	..	16	14	..	79	74	..	93	53	92	56	...	..	99.1	99.2	
6	M	44	72.5	173	3-16-31	10,000	1,500	107	Basal	68	57	62	57	17	16	..	58	50	52	86	56	105	69	97	65	97.8	97.4	Voluntary restricted diet.
6	..	..	73.0	...	3-10-31	10,000	500	73	Basal	68	54	61	57	..	..	..	60	55	60	113	67	113	69	120	71	98.0	97.8	Voluntary restricted diet.
6	..	..	71.0	...	3-27-31	10,000	1,500	135	After breakfast	..	76	71	..	..	..	..	66	64	..	88	60	97	59	...	..	98.3	98.3	Severe cold, restricted diet.
7	M	25	59.0	179	3-16-31	10,000	1,500	61	After breakfast	(67)	60	81	76	19	14	16	68	66	67	114	68	111	71	111	72	98.4	98.6	Slender and weak.
10	M	32	55.5	168	7-14-32	....	100,000	57	After breakfast	(61)	81	77	79	..	..	..	70	62	61	123	60	116	62	120	63	98.6	98.4	Apparently normal.
16	F	33	61.0	174	2-22-33	1,000,000	....	72	Basal	55	65	61	..	18	16	..	66	66	..	83	51	88	57	...	..	97.5	97.4	Apparently normal.
16	..	..	...	...	11-11-32	1,250,000	....	49	After breakfast	..	69	67	69	16	16	18	68	64	68	92	59	98	69	98	67	97.8	97.6	
16	..	..	...	...	7-14-32	....	100,000	70	After breakfast	..	70	65	66	16	12	..	70	66	71	91	62	97	66	97	64	...	98.5	
17	M	24	67.0	186	3-10-33	....	500,000	58	Basal	73	73	73	..	14	13	..	78	74	..	114	67	113	68	...	..	98.1	97.9	Apparently normal.
17	..	..	...	...	2-22-33	500,000	....	71	Basal	73	76	71	..	17	12	..	66	66	..	95	56	115	73	...	..	97.5	97.6	
17	..	..	...	...	7-15-32	....	100,000	65	After breakfast	..	72	67	72	15	..	..	70	61	64	108	65	108	65	112	68	97.6	97.7	
18	M	36	63.0	169	3- 1-33	500,000	....	70	Basal	64	66	66	..	14	15	..	66	64	..	105	65	100	69	...	..	97.5	97.6	Apparently normal.
18	..	..	...	...	7-13-32	....	100,000	25	After breakfast	..	92	84	80	..	..	..	83	74	84	114	69	112	64	114	71	98.6	98.5	
25	M	35	55.5	154	4-15-33	....	250,000	67	Basal	57	57	59	..	19	21	..	64	65	..	128	87	127	83	...	..	97.7	98.0	Japanese, apparently [normal.
25	..	..	...	...	11-15-32	150,000	50,000	69	After lunch	..	86	82	82	..	..	..	77	77	79	138	82	129	87	140	93	98.3	97.8	
30	F	27	59.0	162	11-11-32	500,000	....	50	After breakfast	(55)	72	63	72	..	..	..	84	80	75	125	70	114	73	119	72	98.6	99.0	History of heart trouble.
40	M	29	87.0	179	4- 7-33	150,000	....	44	Basal	82	77	83	..	12	14	..	69	64	..	85	56	105	72	...	..	98.0	97.9	Apparently normal.
40	..	..	...	...	1-27-33	1,000,000	....	73	Basal	82	82	76	..	16	14	..	59	59	..	95	64	115	73	...	..	97.4	97.4	
45	F	15	39.5	158	4- 8-33	150,000	....	69	Basal	51	54	48	..	17	15	..	67	69	..	107	65	107	66	...	..	97.7	97.7	Malnourished.
45	..	..	39.0	...	2- 9-33	....	1,500,000	63	After lunch	..	59	55	..	19	16	..	74	72	..	122	78	112	79	...	..	...	98.5	
51	M	24	70.0	163	4-15-33	....	500,000	79	Basal	70	70	69	69	24	20	17	69	64	67	120	74	109	72	109	72	97.1	97.1	History of heart trouble.
59	M	49	69.0	164	4-14-33	150,000	....	67	Basal	63	56	64	62	18	17	17	52	57	53	97	57	99	59	108	66	98.0	97.7	Apparently malnourished.
60	M	29	70.0	186	4-14-33	150,000	....	69	Basal	74	68	67	66	19	17	18	67	58	58	102	66	102	66	103	66	98.1	97.9	Apparently normal.

of careless leaks through the pneumatic seal of the mask or because of psychic disturbance in the patient while inhaling through the mask. Routine tests for leaks were made by reading the spirometer every minute or two, and by testing

TABLE 2. METABOLIC CHANGES IN IONIZED AIR

Heat Production in Normal Air in Percentage of Harris-Benedict Basal Standards	75–84 (A)	85–94 (B)	95–104 (C)	105–114 (D)	115–124 (E)	125 and Over (F)
Observations Under Basal Conditions						
Average heat production, in % of H.-B. Standards	81.0	91.7	100.2	108.6	121.0	...
Av. change in ionized air in % of H.-B. standards	+12.7	+6.0	−2.4	−4.0	−9.0	...
No. of observations (23 subjects)	3	3	12	8	2	...
No. showing increase (+2 cals. or more)	3	2	2	1	0	...
No. showing no change (within ±2 cals.)	0	1	5	1	0	...
No. showing decrease (−2 cals. or more)	0	0	5	6	2	...
Observations 2 to 4 Hours after Breakfast						
Average heat production, in % of H.-B. basal standards	78.0	92.7	98.9	108.8	118.8	133.4
Av. change in ionized air in percentage of H.-B. standards	+30.0	+19.7	+2.6	−3.4	−6.8	−22.7
No. of observations (35 subjects)	1	3	12	19	11	13
No. showing increase	1	2	6	3	1	0
No. showing no change	0	0	2	4	1	0
No. showing decrease	0	1	4	12	9	13
Observations 3 to 5 Hours after Lunch						
Av. heat production, in % of H.-B. basal standards	...	90.0	98.5	110.9	119.9	134.1
Av. change in ionized air in percentage of H.-B. standards	...	+18.0	+10.5	−0.9	−3.5	−12.9
No. of observations (24 subjects)	...	1	2	8	11	16
No. showing increase	...	1	2	3	2	0
No. showing no change	...	0	0	1	2	0
No. showing decrease	...	0	0	4	7	16

the whole circuit periodically. Any questionable results were discarded. Fortunately most of the subjects were employees of the school and were fairly familiar with the whole procedure.

The blood pressure was determined by the auscultatory method, using a Baumanometer (mercury column) and a bracelet stethoscope. The first clear thumping sound indicated the systolic pressure, and the disappearance of all

sound (fifth phase) was taken as the diastolic criterion, except in two instances in which the fourth auscultatory phase had to be chosen.

RESULTS

The results in this paper are based on 141 experiments with 60 individuals between the ages of 10 and 68 years. Twenty-five were females and 35 males. Of the total number of subjects, 45 were apparently healthy and of the remaining 15, seven were affected by arthritis, one had pulmonary tuberculosis, one hypertension, two were affected by extreme nervousness, one by anemia, and three by malnutrition.

TABLE 3. RESPIRATORY CHANGES IN IONIZED AIR IN RELATION TO METABOLIC CHANGES
(Consolidated Data of Three Series of Observations)

Per Cent Change in Actual Heat Production	+25 or More	+24 to +15	+14 to +5	+4 to −5	−6 to −15	−16 or More
Av. change in heat production (%)...	+32.2	+18.0	+9.0	−1.1	−9.0	−22.3
Av. change in lung ventil. rate (%)...	+23.5	+11.7	+5.9	−0.9	−7.9	−16.3
No. of observations (60 subjects).....	4	3	15	49	42	10
No. showing increase (+2% or more)	4	2	12	20	3	0
No. showing no change (within ±2%)	0	1	0	8	5	0
No. showing decrease (−2% or more)	0	0	3	21	34	10
Av. change in respiration rate (%)...	−13.7	−7.2	−1.5	−2.4	−7.9	−6.8

TABLE 4. CHANGES IN RESPIRATION RATE DURING EXPOSURE TO IONIZED AIR
(Consolidated Results of Three Series of Observations)

Respiration Rate in Normal Air	9–13 (A)	14–19 (B)	20 and Over (C)
Average respiration rate....................	11.5	16.8	21.3
Av. change on exposure to ionized air.........	+2.0 (+17.4%)	−1.4 (−8.0%)	−3.5 (−16.4%)
No. of observations (47 subjects).............	8	62	17
No. showing increase......................	6	8	0
No. showing no change....................	1	7	1
No. showing decrease......................	1	47	16

A summary of individual observations on 15 subjects is given in Table 1 and the average changes among homogeneous groups of subjects are presented in Tables 2 to 7. The significance of the physiologic changes in Table 1 will become more apparent after considering first the general trends. It is sufficient to say at this point that within the limits of the experiments the changes observed with positive and negative ionization were quite similar, regardless of concentration of ions. For this reason the data have been grouped together irrespective of sign or concentration, in establishing the general trend.

The threshold and optimum concentrations of ions for ventilation work will be taken up in another paper.

CHANGES IN METABOLISM

In Table 2 are shown the changes in heat production (*a*) under basal conditions, (*b*) two to four hours after breakfast, and (*c*) three to five hours after lunch. The results are based on a total of 125 sets of observations out of a total of 141 tests; the difference represents the number of questionable observations on new subjects which had to be discarded. The actual metabolism is expressed in percentage of the Harris-Benedict [9, 5] normal basal standards. The average change in heat production on exposure to ionized air is the mean of the individual changes which were classified according to the initial heat production at the *steady state* (before ionization).

Referring to the observations under basal conditions (Table 2) the minimum change during exposure to ionized air occurred in Group *C* in which the heat production before ionization was within $\pm$ 5 per cent of the normal standards. A definite increase took place in five of the six instances of Groups *A* and *B* in which the initial heat production was less than 95 per cent of the normal, the average increase being 12.7 per cent and 6.0 per cent respectively for the two groups. Reversely, the basal metabolism rate decreased in eight of the ten instances in which it was more than 105 per cent of the normal (Groups *D* and *E*), the average decrease being 4.0 per cent and 9.0 per cent for Groups *D* and *E* respectively.

In Group *A* showing an initial basal metabolism less than 85 per cent of the normal, there were two men and one woman. One of the men (See Table 1, Subject 6), an employee of the School was under a voluntary restricted diet, losing weight rapidly. Two observations about three weeks apart showed basal metabolic rates of 78 and 80 per cent of the normal in spite of a small breakfast on both occasions. The second man was a professional subject apparently in a malnourished condition at the time of the experiments. He was studied three times and found to have a consistently subnormal metabolic rate. The woman in the group, one of the author's (L. C. B.), showed no abnormality to account for the spontaneous low metabolism. On four other occasions her metabolic rate was well within the normal range.

There is also no obvious reason for the comparatively high basal metabolism in the two subjects of Group *E* except that one of them was very small, weighing only 45.5 kg. Both were women about 60 years old, affected by rheumatic conditions.

In the series of observations made after the subjects had eaten breakfast or lunch, the changes in heat production following exposure to ionized air are quite concordant with those of the basal series (Table 2). Under the conditions of the experiments, breakfast in itself increased the heat production in normal air on an average of about 10 per cent above the Harris-Benedict basal standards, and a light lunch increased it by about 20 per cent. In other words the average normal metabolic rate before ionization was about 110 per cent of the basal standard in the experiments after breakfast, and about 120 per cent in the experiments after lunch. Correspondingly, little change took place in ionized air in Groups *C* and *D* after breakfast, and in Groups *D* and *E* after lunch. In the remaining groups the metabolism was appreciably affected.

The five subjects in Groups *A* and *B* showing subnormal initial metabolic

[9] A Biometric Study of Basal Metabolism in Man, by J. A. Harris and F. G. Benedict, Carnegie Inst. Washington Pub., No. 279, 1919.

rates despite the stimulating action of food were affected by such conditions as anemia, malnutrition or a run-down condition.

In instances in which the observations were continued after the ionization period, the metabolic rate tended to return to its pre-ionization level, as can be seen in Table 1.

RESPIRATORY CHANGES

As shown in Table 3, the changes in metabolism were effected largely by variations in the lung ventilation rate rather than in the composition of the expired air. Variations in the pulmonary ventilation rate were in turn affected entirely by variations in depth rather than in rate of respiration. As a matter of fact, the respiration rate decreased on an average (Table 3), despite the increases in pulmonary ventilation and metabolism.

The changes in the respiration rate during the ionization period depended largely upon the initial rate in normal air (Table 4). Although as a rule the basal rate was the lowest, and that after lunch the highest of the three series of observations, the differences were too small to justify treating each series separately, as was done with the metabolism.

In some of the experiments, the respiration rate was counted only during the metabolism period when the subjects were breathing through the inhalator mask. These data are not included in Table 4 because the respiration as a rule was slower and deeper when the mask was worn than without it.

Of a total of 87 observations, in 62 (or 71 per cent) the respiration rate in normal air was between 14 and 19, averaging 16.8 per minute (Table 4). The average change in this group (*B*) on exposure to ionized air was −1.4 breaths per minute. In Group *A* showing initial rates between 9 and 13, the rate increased on an average of 2.0 breaths a minute during the ionization period, whereas in Group *C*, having a respiration rate of 20 or more, the rate decreased by 3.5 breaths a minute. The percentage change in Groups *A* and *C* is +17.4 per cent, and −16.4 per cent respectively.

The characteristics of respiration in breathing polarized air will be discussed in another paper. It may be of interest to note at this time that the respiration rate was more regular than in normal air, especially in subjects with irregular breathing, and that the depth of respiration was considerably increased.

CHANGES IN PULSE RATE

The pulse rate in normal air during the steady state varied widely among the subjects, particularly among the women. The maximum variation in the groups was from 50 to 96. As in the case of respiration rate, the basal pulse was somewhat lower than the rate after breakfast, and the rate after lunch was the highest; the changes in the three series, however, were sufficiently concordant to allow their being treated together as in Table 5.

The most popular pulse rate was between 66 and 75 in subjects under 40 years of age, and 65 or less (55-65) in subjects over 40 years. In these two groups there was a small decrease on exposure to ionized air which might be attributed to prolonged rest. The pulse rate in ionized air increased slightly in subjects under 40 years of age, showing initial rates of 65 or less, and decreased appreciably in the majority of subjects who had initial rates of 76 or more regardless of age.

Taken by themselves, these changes in pulse rate may have little physiologic significance except that they are consistent with those in metabolism, respiration, and blood pressure. After the ionization period, the pulse rate, like the metabolic rate, tended to return to its preionization level.

CHANGES IN BLOOD PRESSURE

The changes in blood pressure depended considerably on the age of the subject, as shown in Tables 6 and 7. Among persons under 25 years of age the normal (lying) systolic blood pressure of the majority during the steady

TABLE 5. CHANGES IN PULSE RATE DURING EXPOSURE TO IONIZED AIR
(Consolidated Data of Three Series of Observations)

Pulse Rate in Normal Air	65 or Less	66–75	76 and Over
Age Under 40 Years			
Average pulse rate	61.4	69.8	81.6
Average change in ionized air	+1.8	−2.3	−7.1
No. of observations (44 subjects)	21	46	39
No. showing increase (+2 or more)	9	5	0
No. showing no change (within ±2)	8	12	7
No. showing decrease (−2 or more)	4	29	32
Age 40 Years and Over			
Average pulse rate	63.3	69.7	82.7
Average change in ionized air	−0.2	−4.3	−7.8
No. of observations (16 subjects)	19	10	6
No. showing increase	8	0	0
No. showing no change	4	1	0
No. showing decrease	7	9	6

state ranged between 90 and 114, and the change in ionized air was slight (+1.8 mm Hg). In subjects of the same age group who showed initial systolic pressures between 115 and 129, the pressure was reduced by an average of 11.0 mm during exposures to ionized air.

Of the three age groups the oldest had the highest blood pressure, and seemed to be affected most during the ionization period. On an average, the systolic pressure in this group increased 12.2 mm when the preionization pressure was less than 90 mm, and decreased 13.1 mm when the initial pressure was 130 mm or more.

It may be worth noting that in the pressure group of 115 to 119 mm (Table 6), persons under 25 years received a significant decrease, whereas those over 40 were but little affected by ionization. The blood pressure readings on the single subject suffering from hypertension (245 mm Hg) have been omitted from Table 6 in order to avoid distortion of the general trend.

Changes in diastolic pressure during the ionization period are presented in Table 7.

As in the case of metabolism, respiration, and pulse rate, the blood pressure also showed a tendency to return to its preionization value after the ionizer was turned off (See Table 1).

CHANGES IN BODY TEMPERATURE

The body (oral) temperature varied but little during the ionization period, despite the substantial changes in metabolism (Table 1).

PHYSIOLOGIC CHANGES IN CONTROL EXPERIMENTS

In Table 8 are presented the results of control experiments which were carried out under the conditions of the three main series, except for the omission

TABLE 6. CHANGES IN SYSTOLIC BLOOD PRESSURE DURING EXPOSURE TO IONIZED AIR
(Consolidated Results of Three Series of Observations)

Systolic Blood Pressure in Normal Air (mm Hg)	Under 90	90–114	115–129	130 and Over
Age Under 25 Years				
Average pressure	(88.0)	106.4	120.9	...
Av. change in ionized air (mm Hg)	(+1.0)	+1.8	−11.0	...
No. of observations (8 subjects)	1	23	8	...
No. showing increase (+2 mm. or more)	0	7	0	...
No. showing no change (within ±2 mm)	1	13	0	...
No. showing decrease (−2 mm or more)	0	3	8	...
Age 25 to 39 Years				
Average pressure	83.9	102.9	120.8	138.0
Av. change in ionized air	+9.8	+0.9	−3.7	−6.7
No. of observations (36 subjects)	9	48	11	6
No. showing increase	9	20	2	0
No. showing no change	0	12	3	1
No. showing decrease	0	16	6	5
Age 40 Years and Over				
Average pressure	87.7	108.2	120.2	143.5
Av. change in ionized air	+12.2	+1.6	±0.0	−13.1
No. of observations (15 subjects)	4	14	8	8
No. showing increase	4	6	2	0
No. showing no change	0	1	3	3
No. showing decrease	0	7	3	5

of the ionization period. The data shown, except those for the metabolism, are averages of the values recorded after resting periods of one, two, and three hours approximately. The subjects knew nothing of the purpose of these tests, and they probably assumed that the air was being ionized as usual.

The variations in the physiologic functions under basal conditions are too small to be of any significance (Table 8). In the experiments after breakfast and after lunch the variation is greater than in the basal series, but the direction of the change is irregular and inconsistent with that in Tables 1 to 7. Judging from this relatively small number of control tests, the average changes as shown at the right of Table 8 appear to be too small to account for the alterations observed following the administration of ionized air.

Subjective Sensations in Ionized Air

In the first 17 experiments the ionizer produced a distinct buzzing sound which aroused apprehension, particularly in three new subjects. Judging from their remarks, they expected something unusual to happen and were rather disappointed because nothing occurred. The condition was probably aggravated by questioning the subjects concerning transitional changes in their sensations immediately after turning the ionizer on, a practice which was soon discontinued. Needless to say, all questionable observations have been discarded.

Table 7. Changes in Diastolic Pressure During Exposure to Ionized Air
(Consolidated Results of Three Series of Observations)

Diastolic Blood Pressure in Normal Air (mm Hg)	Under 60	60–69	70–79	80 and Over
Age Under 25 Years				
Average pressure	56.4	63.5	74.3	...
Av. change in ionized air (mm Hg)	+8.1	+2.3	−3.0	...
No. of observations (8 subjects)	10	18	4	...
No. showing increase (+2 mm or more)	9	10	1	...
No. showing no change (within ±2 mm)	1	7	1	...
No. showing decrease (−2 mm or more)	0	1	2	...
Age 25 to 39 Years				
Average pressure	54.2	64.2	74.4	85.2
Av. change in ionized air	+5.6	+1.0	+0.5	±0.0
No. of observations (36 subjects)	32	26	11	5
No. showing increase	23	11	5	2
No. showing no change	5	7	4	1
No. showing decrease	4	8	2	2
Age 40 Years and Over				
Average pressure	56.2	65.3	74.9	84.1
Av. change in ionized air	+6.7	+2.1	+2.6	−5.1
No. of observations (15 subjects)	6	10	8	10
No. showing increase	5	6	4	0
No. showing no change	1	1	3	2
No. showing decrease	0	3	1	8

In most instances, subjects from outside the School knew little or nothing about the ionization part of the experiment, and only a few in the whole group knew whether the ions were positive, negative, or mixtures of the two.

Table 9 shows the sensations recorded at the close of the tests. Approximately in 15 to 30 per cent of the observations there was no subjective change whatever, and the subjects were unconcerned as to whether the ionization was positive, negative, or a mixture.

In experiments with positive ions, the most definite sensations which were reported with sufficient regularity by a certain group of subjects (evidently

TABLE 8. PHYSIOLOGIC CHANGES IN CONTROL EXPERIMENTS
(No Artificial Ionization)

Resting Period in Normal Air—Hours	1	2	3	1	2	3	1	2	3	1	2	3	Average Change: 1st to 2nd Hour	Average Change: 2nd to 3rd Hour	Average Change: 1st to 3rd Hour
Observations Under Basal Conditions	Subject #2 Female, 27 Years			Subject #7 Male, 25 Years			Subject #8 Male, 46 Years								
Actual heat production, calories per hour	58.7	....	58.2	68.6	....	....	65.7	66.0	65.4				+0.3	−0.6	−0.4
Heat production in % of H.-B.standards	107	....	106	102	....	....	97	97	96				±0.0	−1.0	−1.0
Respirations per minute	14	13	14	18	17	17	13	14	12				−0.3	−0.3	−0.7
Pulse rate	68	68	68	64	62	65	63	63	64				−0.7	+1.3	+0.7
Systolic blood pressure, mm Hg	85	82	85	112	109	112	118	120	118				−1.3	+1.3	±0.0
Diastolic blood pressure, mm Hg	53	49	50	72	69	71	69	70	70				−2.0	+1.0	−1.0
Oral temperature, F	99.1	99.0	99.0	98.3	98.3	98.1	98.0	97.8	97.8				−0.1	−0.1	−0.2
Observations Two to Four Hours after Breakfast	Subject #11 Male, 44 Years			Subject #13 Male, 32 Years			Subject #59 Male, 49 Years			Subject #60 Male, 29 Years					
Actual heat production	68.4	71.8	....	75.9	73.4	....	60.1	....	63.0	71.2	....	70.1	+0.5	....	+0.9
Heat production in % of H.-B.standards	118	124	....	110	106	....	96	....	100	96	....	94	+1.0	....	+1.0
Respirations per minute	....	....	....	17	18	....	16.5	15	16	18.5	18	18	−0.3	+0.5	−0.5
Pulse rate	60	62	....	66	63	....	65	62	64	61	61	59	−1.0	±0.0	−1.5
Systolic blood pressure	87	89	....	113	109	....	99	97	103	99	96	103	−1.8	+6.5	+4.0
Diastolic blood pressure	58	60	....	64	64	....	54	57	59	58	59	60	+1.5	+1.5	+3.5
Oral temperature	98.5	98.4	....	98.4	98.0	....	98.1	98.1	98.2	97.7	97.7	97.6	−0.1	±0.0	±0.0
Observations Three to Five Hours after Lunch	Subject #2			Subject #59			Subject #60								
Actual heat production	67.0	62.1	67.3	69.8	74.2	74.7	77.6	79.8	76.4				+0.6	+0.8	+1.3
Heat production in % of H.-B.standards	122	113	123	111	118	119	105	108	103				+0.3	+2.0	+2.3
Respirations per minute	20	21	18	18	18	18	16	18	18				+1.0	−1.0	±0.0
Pulse rate	78	75	74	66	65	67	63	63	65				−1.3	+1.0	−0.3
Systolic blood pressure	89	85	88	100	109	107	101	98	102				+0.7	+1.7	+2.3
Diastolic blood pressure	51	50	50	59	65	66	57	60	64				+2.7	+1.7	+4.3
Oral temperature	99.0	99.0	99.0	98.3	98.2	98.2	97.9	97.9	97.6				±0.0	−0.1	−0.1

sensitive to positive ions) to make the data valid, are dryness and irritation in the nose and throat, and frontal headaches (Table 9). This is entirely in accord with Dessauer's experience.[3] The apparent superiority in freshness and stimulating quality of mixtures of positive and negative ions (the former predominating) was most noticeable in the first 17 experiments in which there was a trace of ozone in the air, although the concentration was below the threshold of detection by the potassium-iodide starch method. Strips of white blotting paper dipped into a fresh iodide-starch solution turned in color on exposure to the stream of ionized air before the latter mixed with the main

Table 9. Subjective Sensations in Ionized Air
(Exposure 1 to 2 hours.)

Sensations	Per cent of Total Observations					
	Positive Ions	Negative Ions	Positive and Negative	Positive Ions	Negative Ions	Positive and Negative
No change	21.7	27.6	13.5	21.7	27.6	13.5
Fresher	1.3	10.3	21.7*			
General cooling effect	5.4	12.9	0.0			
Invigorating, stimulating	10.8	9.5	32.4*	40.5	71.5	73.0
Relaxation	8.1	21.6	2.7			
Sleepiness	9.5	12.9	10.8			
Increased appetite	5.4	4.3	5.4			
Dryness and irritation of nose	13.5	0.0	5.4			
Headache	13.5	0.9	5.4	37.8	0.9	13.5
Restlessness	2.7	0.0	0.0			
Probable diuresis	8.1	0.0	2.7			
Total number of observations	74	116	37	100.0%	100.0%	100.0%

* Attributed largely to trace of ozone in the air.

ventilating current, but the strips dried up without changing color on exposure to the breathing zone of the room.

In all subsequent experiments with improved ionizers (producing no ozone) this superiority did not persist. In a special series of tests with ozone concentrations distinctly perceptible to the nose, the sensation was that of irritation rather than freshness.

Negative ionization had a tendency to produce a cooling effect on the body, and this was most apparent in the summer tests when the skin of the subjects was more or less moist. The most popular sensation was one of relaxation and, in extreme cases, sleepiness (Table 9). In many instances these effects were discernible a few minutes after turning on the ionizer, and they occurred mainly in subjects having a comparatively high blood pressure, a high pulse rate, or a high metabolism. As a rule relaxation was concomitant with a fall in blood pressure, the pulse rate, and metabolism. Conversely, sensations of stimulation or invigoration were accompanied by an acceleration in these physiologic functions in subnormal cases.

Summing up the results in Table 9, it can be seen that 40.5 per cent of the sensations recorded after exposure to positive ions for one hour or more were of an agreeable nature, and 37.8 per cent were somewhat disagreeable. In negatively ionized air, on the other hand, 71.5 per cent of the sensations were of a desirable character, and there were almost no unpleasant effects. Despite this apparent favoritism to negative ionization, only 11 of a total of 45 subjects exposed to it for one hour or more expressed a strong liking for it and felt that they were benefited. All 11 were from the abnormal group of 15 subjects whose condition was described earlier in the paper.

Table 10 gives the results of a special series of tests in which interested visitors of the School during the past two years reported their sensations after

TABLE 10. PRIMARY SENSATIONS IN IONIZED AIR
(Special tests. Exposure 5 to 20 minutes)

Sensations	Per cent of Total Observations			
	Positive Ions	Negative Ions	Positive Ions	Negative Ions
No difference	5.9	0.0	5.9	0.0
Fresher	32.3	45.5		
Easier to breathe	0.0	15.1		
General cooling effect	2.9	18.2		
Tingling in skin and nose	8.8	9.1	46.9	100.0
Invigorating, stimulating	2.9	3.0		
Relaxation	0.0	9.1		
Difficult to breathe	11.8	0.0		
Dryness and irritation in nose	14.8	0.0	47.2	0.0
Headache	8.8	0.0		
Depressing; dullness in head	11.8	0.0		
Total number of observations	34	33	100.0%	100.0%

the ionizer was turned on and off two or three times, and after short exposures. All persons in this group were fully aware that the air was being ionized, but most of them did not know just what polarity they were getting until after the test was over.

According to these data there seems to be an appreciable freshness in ionized air, particularly in the case of negative ionization, but as in the main experiments, the preference was not strong to suggest a definite improvement in the condition of the air, except in a few instances.

SUMMARY

The influence of ionized air (small ions) upon total metabolism, respiration, pulse rate, blood pressure, and body temperature, was studied on human subjects lying on cots, (*a*) under basal conditions, (*b*) two to four hours after breakfast, and (*c*) three to five hours after a light lunch. A group of 60 persons in a total of 141 experiments were exposed for a period of one hour or more to air containing from 5,000 to 1,500,000 ions per cubic centimeter (normal air contains from 50 to 800 small ions per cubic centimeter), after a preliminary resting period of one to two hours in normal air.

Under the experimental conditions ionized air appeared to exert a normalizing influence upon the organism by accelerating the physiologic processes in instances in which these processes were below the normal range of the majority of subjects, and reversely, by decreasing the physiologic activity in cases in which the functions were above the normal range. Following the ionization period there was a reversion of the functions toward their steady state prior to the ionization period. This indication, being of a somewhat surprising nature, requires verification by further experiments.

Although the physiologic response to positive and negative ions did not seem to differ greatly, certain differences were observed in the sensations produced by the two kinds of ions; positive ionization resulted in headaches and irritation in the nose and throat in some cases, whereas negative ionization predisposed to relaxation and other sensations of a desirable character. In a few instances these effects were discernible a few minutes after turning on the ionizer.

A minority of the subjects reported an appreciable freshness in ionized air, particularly in the case of negative ionization, but the preference was not strong enough to suggest a striking improvement in the condition of the air except in the case of abnormal persons. Out of the 45 subjects exposed to negatively ionized air for an hour or more, only 11 expressed a strong liking for it and felt that they were benefited. All 11 were from an abnormal group of 15 subjects. Even in such special cases, however, the effects of continuous exposure are unknown and not much is known about the optimum concentrations. The problem is now being studied in the laboratories of this School and in the wards of nearby hospitals.

DISCUSSION

Dr. C. A. Mills [10] (Written): The work being done in the Harvard laboratory seems to me to be of great importance, and to offer findings of practical value. I wonder if, perhaps, it might not be advisable to keep the number of ionized particles, both positive and negative, closer to the level of actual living conditions. The observations here reported were made at degrees of ionization far in excess of those met with in every-day life, or even with our severe storm changes in weather. I am hoping that some day Dr. Yaglou will find just what it is in our severe changes of weather which so alters man's physiologic functions,—why we are so irritable and restless with rapidly rising temperature and falling pressure, and so much more calm and efficient when the temperature is falling and pressure rising. Under the latter set of conditions we speak of the cool invigorating air as being like wine, and we feel equal to any task. These changes are not merely psychic, but must be based on metabolic differences. I suspect the state of atmospheric ionization to play an important rôle and thus am much interested in the work being carried on.

J. N. Hadjisky: How expensive and how large is the apparatus used for generation of ionized air?

What is the cost of operation of air ionizing apparatus for a given delivery, such as supplying the required ionized air for 10 persons per hour, or for supplying a space of 1,000 cu ft to be maintained at required ionized air concentration?

C.-E. A. Winslow: The claims of Dessauer and his associates as to the effects of ionized air seem to the physiologist so miraculous as to arouse serious doubt of their validity. Nevertheless the problem is so important that I made a special trip to

[10] Professor of Experimental Medicine, University of Cincinnati.

Frankfort last summer to study the techniques involved and we have obtained for the Pierce Laboratory of Hygiene at New Haven a duplicate of the Dessauer apparatus.

Professor Yaglou has presented his results with characteristic caution, but these results certainly appear to confirm the major claims of the German workers. We should all be encouraged to proceed with the study of this unsuspected atmospheric influence. It is clear, as Professor Yaglou has pointed out, that the subject is one for further research and that premature commercial exploitation would be unfortunate. The Harvard laboratories have, however, given us a lead of the very first importance.

C. P. YAGLOU: Ionizing apparatus is quite inexpensive in first cost, and the cost of operation is less than that of an ordinary electric light bulb.

No. 966

MEASUREMENT OF THE FLOW OF AIR THROUGH REGISTERS AND GRILLES

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This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the *Ventilating Contractors Employers Association* of Chicago and conducted at Armour Institute of Technology of Chicago

THE results obtained in the laboratories of Armour Institute of Technology from tests conducted to determine the most accurate and simple method of measuring the air flow through registers and grilles have been presented to the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS at two previous meetings. As it appeared that the only instrument that was practical for use in this manner was the anemometer, the work resolved itself into a study of that instrument and its characteristics. It must be clearly understood that there was no expectation of developing a precise laboratory method, but rather a method applicable to field tests where errors not to exceed 5 per cent may be considered reasonable.

The investigations were conducted with both supply and exhaust grilles. The set-up consisted, briefly, of a well constructed system of ducts so arranged that the same air that passed through the grilles was also drawn or forced through a straight round section of duct of relatively small cross-section to insure rather high velocities. Pitot tube traverses taken in two directions across this pipe, were used as the basis for comparison with anemometer traverses taken by various means at the face of the grille. This test set-up, as arranged for supply grilles, is shown in Fig. 1.

To prevent confusion the discussion will be confined to supply grilles.

In the first paper [1] a new formula was developed, namely:

$$\text{cfm} = \frac{C_v V(A + a)}{2}$$

in which

cfm = volume of air in cubic feet per minute

V = average indicated velocity obtained by an anemometer traverse taken in contact with grille face (after applying the usual instrument correction)

A = gross area of grille, square feet

a = net free area, square feet

C_v = coefficient obtained by experiment

† Associate Professor of Experimental Engineering, Armour Institute of Technology.

[1] The Measurement of the Flow of Air Through Registers and Grilles, by L. E. Davies, A. S. H. V. E. TRANSACTIONS, Vol. 36, 1930, p. 201.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Hotel Statler, Detroit, Mich., June, 1933.

The virtue of this formula was that this coefficient was very close to unity, so for many practical purposes, it could be omitted entirely.

The observations upon which this formula was based were of a rather limited character, being confined to thin metal grilles of a simple lattice work design, except for one observation with a piece of expanded metal lath. One type of anemometer was used throughout the early experiments, and most of the observations were taken with fairly satisfactory types of approach ducts. Since that time an attempt has been made to determine how to apply this

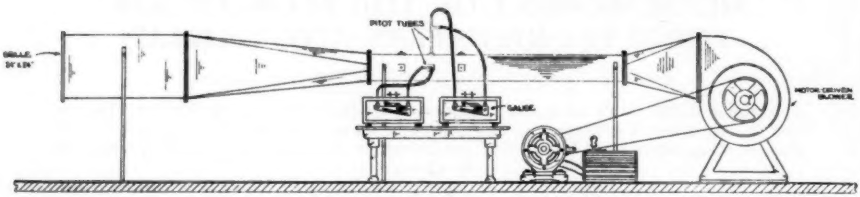


FIG. 1. TEST SET-UP FOR SUPPLY GRILLES

formula to the many variable conditions encountered in practice. These conditions include:

1. Different types of anemometers.
2. Grilles of ornamental design.
3. Registers of various special and unique design.
4. Unit heater cores and unit ventilators.
5. Various types of approach ducts.
6. Grilles of various dimensions.
7. Lattice work grilles in which the frets are relatively large.
8. Grilles backed by screens, heater cores, adjustable dampers, etc.

Some of these cases were discussed in a previous paper,[2] particularly the use of the different types of anemometers. The most important disclosure brought out in this second report, however, was the fact that when the formula was applied to two special grilles of an ornamental plaster design, the results were very unsatisfactory, the error in one case amounting to 20 per cent while in the other it reached a peak value of 45 per cent. Recently considerable time has been devoted to an attempt to explain this phenomena with results that will be discussed in detail elsewhere in this report.

In order to have a basis for comparison a certain set of conditions will be assumed, and on the basis of an earlier report these conditions will be designated as normal. The specifications for these normal conditions are given in Table 1.

Now it is apparent that if any one of these conditions is changed the results obtained may be in error. For example, if a condition existed in the approach duct which caused the air to be discharged at a pronounced angle it might conceivably cause the anemometer to read low. If this were a fact, and the normal procedure and normal formula were used, the calculated volume of air

[2] The Measurement of the Flow of Air Through Registers and Grilles, by L. E. Davies, A. S. H. V. E. TRANSACTIONS, Vol. 37, 1931, p. 619.

would be lower than the true volume by an amount proportional to the error in the anemometer reading. This error might be corrected for either by some modification in the standard procedure, or by the application of special coefficients or correction factors.

Considering the number of variable factors it might appear then that it would be necessary to use a veritable book of coefficients and special instructions to deal effectively with the various cases. Fortunately, this is not true.

Table 1. Specifications for Normal Conditions in Using Anemometer to Measure Air Discharged from a Grille

Type of grille—Plain lattice work of thin metal.
Free area 30 per cent to 90 per cent
Overall dimension, feet—2x2
Width of fret—¼ in.
Thickness—1/16 in.
Type of approach—Straight duct full size of grille, free from elbows, dampers, splits or other forms of obstruction.
Character of discharge—Straight forward—uniform distribution.
Character of anemometer traverse—16 spot readings taken at centers of 6 in. squares, anemometer in contact with grille face.
Type of anemometer—4 in.—200 ft to 3000 fpm capacity.

Formula to be used—$\text{cfm} = \dfrac{C_v V(A+a)}{2}$

Values of C recommended.

Air Velocity, fpm	C_v
150	0.952
200	0.957
300	0.967
400	0.977
500	0.985
600	0.992
700	0.998
800	1.000

Several of the factors are of little consequence and others can be handled by the application of a few simple rules of procedure.

Let us now consider some of these factors and their effect on the accuracy of the results obtained when the normal procedure and normal formula and coefficients are used.

Different Types of Grilles

Considering, first, the effect of changes in the design of the grille itself, Fig. 2 shows some of the grilles that have been tested and found to give results identical with those obtained with the plain lattice work type. It will be noted that the frets in the different types differ considerably. For example, *f* has the small round wire, *c* and *d* thin flat strips half an inch deep, *e* diamond shaped, *a* rather deep tapered frets, etc. In addition to those illustrated several others of similar character have been used, including the lattice work designs with various widths of frets under ⅜ in., and various percentages of free area. Tests were made using ornamental iron and plaster designs where the figures in the design were all of relatively small size. Tests also were made using grilles of various overall dimensions and proportions, from approximately 9 x 12 in. and 5 x 40 in. to the 24 in. square that was used in most of the early

work. So far as the design and size of grilles is concerned, the conclusions are that the normal procedure can be followed within the following limitations:

1. Size—anything except possibly those of extremely small dimensions.
2. Percentage of free area—no limitation.
3. Design—either plain or ornamental.
4. Depth of grille—any depth not to exceed approximately 10 times the minimum distance between frets.
5. Width of frets—not to exceed ⅜ in.

It should be noted that throughout this paper the term *grille* will be used in a rather broad sense so that it may refer to anything having the general character of a grille, such for example, as radiator cores. Thus, in Item 4, are called to mind certain heater cores of the tube and fin type, where the depth is several inches, and the distance between fins only a fraction of an inch. In such cases the resistance in the long narrow passages has the effect of a much smaller free area than the actual measured value. Hence, if the standard procedure is followed, the calculated air flow may be some 10 to 15 per cent too high, even when the velocity distribution is uniform. With an irregular velocity distribution such as is obtained when the propeller or disc fan is used, the error may be much greater.

In some instances grilles of the usual types are backed by ordinary wire screen. Tests were made to determine how to handle these situations. The standard procedure can be followed exactly except that in determining the free area both the grille and the screen must be considered. The percentage of free area of the combination, where they are in reasonably close contact, can be taken as the product of the two individual values. Thus, for example, imagine a plain lattice work grille, 4 square feet in area and with 60 per cent free area backed by an 8 mesh screen having 70 per cent free area. The free area *a* to be used in the formula would be:

$$a = 4 \times 0.6 \times 0.7 = 4 \times 0.42 = 1.68 \text{ sq ft}$$

If the screen is several inches removed from the grille proper, it will fall into a class to be discussed later.

GRILLES REQUIRING SPECIAL CONSIDERATION

Certain types of grilles (using the word again in its broad sense) are of such unique design that there naturally exists in the mind of the operator considerable doubt as to what should be taken as the free area. Examples of some of these are shown in the sketches *a, b, c, d,* of Fig. 3, and the photograph, Fig. 4.

Sketch *a* in Fig. 3 illustrates a door louver similar to the one used by Professors Larson and Nelson in their Investigation of Air Outlets,[3] and Sketch *b* is a grille of the same general character as the one illustrated in *d* of Fig. 2, except that the outer edges are turned down about 30 deg.

The general rules in all these cases is to take for free area the ***minimum*** cross sectional area at any point in the air stream. Where the frets are at

[3] Investigation of Air Outlets in Class Room Ventilation, by G. L. Larson, D. W. Nelson and R. W. Kubasta, A. S. H. V. E. TRANSACTIONS, Vol. 38, 1932, p. 463.

an angle this area will be the area between them measured normal to their surface (distance *b* on sketches). As an example of how this rule is applied, consider the door louver illustrated. The anemometer traverse should be made across the face of the louvers in the usual manner, the upper strip through which no air can issue being, of course, not considered as part of the face. The gross area, *A*, in the formula should always be taken as that area over

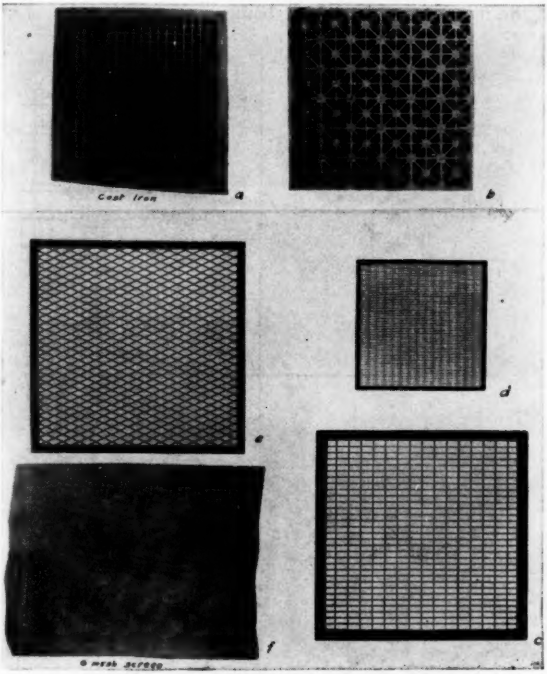


Fig. 2. Examples of Grilles Which Do Not Require Special Treatment

which the traverse is taken, in this case 5 sq ft. The free area would then be taken as

$$a = 2\tfrac{1}{2} \times (2 \cos 45 \text{ deg} - 16 \times \tfrac{1}{4} \times 1/12) = 2\tfrac{1}{2} \times 1.081 = 2.702 \text{ sq ft}$$

The same result would be obtained if the free area at the face were simply calculated and multiplied by the cosine of the angle of the louvers. Or many will prefer to caliper directly the minimum distance between the louvers. The actual method of calculation for free area is optional but in cases of this kind the problem must be studied carefully for it is easy to make an error.

When Professors Larson and Nelson ran their anemometer calibration tests on the door, they found it necessary to use a coefficient of from 0.78 to 0.85 in the $\frac{C_v V(A + a)}{2}$ formula. They used the free area measured at the face. They furnished complete copies of the original data, and when recalculated by the method here described the average coefficient obtained from thirteen runs differed from those which have been recommended by only slightly over 1 per cent, the maximum discrepancy being 5 per cent, obtained at a very low

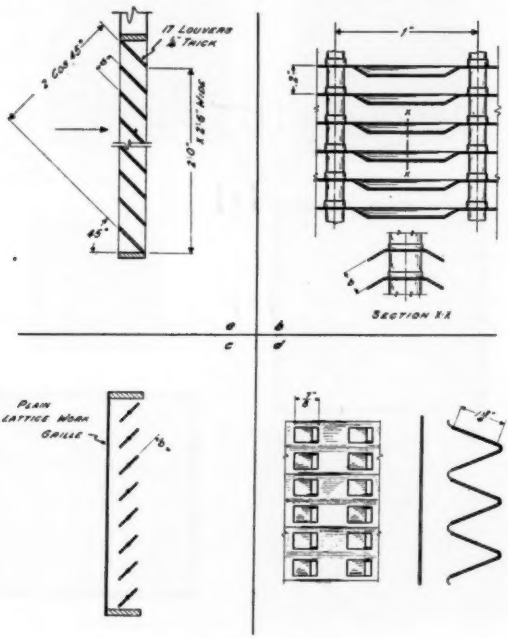


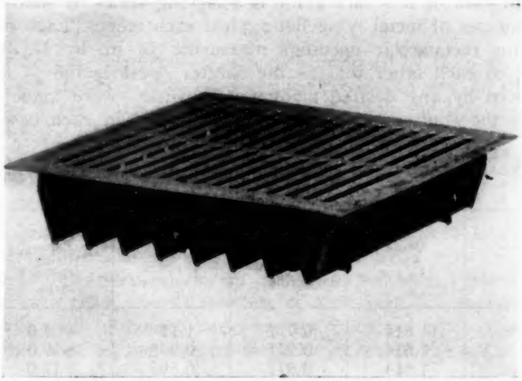
FIG. 3. SPECIAL TYPES OF GRILLES

velocity. These results are regarded as of particular value because of the fact that they were obtained by a different experimenter and with different apparatus.

Another application of the same principle is with the grille shown in Sketch *b,* Fig. 3. As already stated, this is identically the same as the one shown in photograph *d,* Fig. 2, except for the turned down edges. These two types are classed by their maker as straight fin, and angle type fin respectively. The manufacturer lists these grilles as having 84.3 per cent free area, and careful measurement on the straight fin type was made to verify this figure. When traverses were made on the straight fin type by the normal method the results obtained were very satisfactory. With the angle type fins, however, the calculated quantities were some 6 per cent to 7 per cent high. A careful check

on this type showed that when the free area was calculated by taking the minimum distance between the fins (at the point where they slope downward, *b* on sketch) it was found to amount to only 76 per cent of the gross area. Using this smaller figure in the formula the error in the anemometer traverse was reduced to less than 2 per cent.

The sketches, *c* and *d*, of Fig. 3, illustrate two registers with which in-



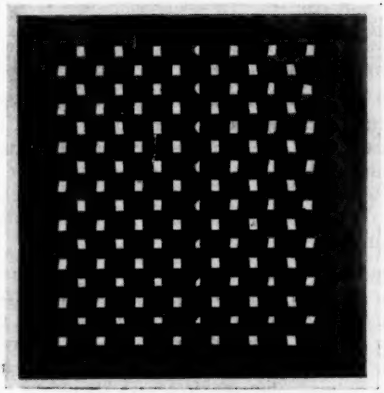


FIG. 4. GRILLE OF UNUSUAL DESIGN

teresting results were obtained. The first of these was a plain lattice work grille, behind which was a set of dampers. Runs were made in the usual way with the dampers set for various angles. The values of *A* and *a* in the formula were first taken directly from the grille face, the presence of the dampers being ignored. It was found that accurate results were obtained up

to the point where the minimum area through the dampers (calculated by the same means used with the louvers already discussed) became less than the free area of the grille itself. Beyond that point the calculated results were too high unless this minimum free area through the dampers was used for free area in the formula.

The second of these registers is of special interest because of its unusual design. Its face may be either the usual lattice work or the bar type shown in Fig. 4. In back of this face grille is a zig-zag series of shutters consisting of two thicknesses of metal lying flat against each other. Each of these metal strips contains rectangular openings measuring ⅞ in. by 1¾ in. so placed with respect to each other that as the shutter operates the ⅞ in. dimension can be reduced by any desired amount. Traverses were made in the usual manner with the shutters in various positions, and in each case calculations

TABLE 2. RESULTS OBTAINED FROM TRAVERSES MADE ON REGISTER SHOWN IN FIGS. 4 AND 3D

Position of Shutters	Gross Area Sq Ft	Net Free Area Through Grille	Net Free Area Through Shutters	Per Cent Error Using Grille Area	Per Cent Error Using Shutter Area
Full open	1.614	0.977	1.19	+ 7.0	+17.5
¼ closed	1.614	0.977	0.935	+ 9.0	+ 7.0
½ closed	1.614	0.977	0.595	+12.0	− 4.7
¾ closed	1.614	0.977	0.2975	+32.0	− 2.5

were made in two ways, first, by using the free area of the grille and secondly, by using the area through the shutters. The results are shown in Table 2.

Here again it will be seen that when the shutters were wide open the greatest accuracy was obtained by using the grille area, which is smaller than the shutter area, whereas when the shutters were partly closed so as to reduce their area to a figure smaller than the free area of the grille, the greatest accuracy was obtained by using the shutter area. The results were somewhat less consistent than those obtained with plain grilles but this was to be expected under the circumstances.

Other observations taken with one grille mounted two or three inches in front of another, or in front of a heater core, indicate that the rule discussed here may be taken as a general one. That is, when a grille has behind it, within a few inches distance, any obstruction of the general character of those described, the gross areas should be measured at the outer face where the anemometer traverse is made, but the free area should be taken at that point in the air stream at which it is a minimum.

The fact has been mentioned that when runs were made on two ornamental plaster grilles by the usual method some very large errors were found. No explanation for this phenomena was offered in the earlier paper but further thought and experiment since that time have produced gratifying results.

In order to find the explanation for the inconsistent results obtained with these grilles it was necessary to analyze them carefully in order to determine

in what manner they differed from the other grilles. This analysis shows the following characteristics:

1. The grilles were quite thick.
2. The designs were of an ornate character.
3. The surfaces were rough and not all in one plane.
4. The outer edges were rounded off.
5. The details of the design were large, that is, large irregular frets and large interstices.

The first four of these factors were quickly eliminated by the fact that satisfactory results were obtained with other grilles possessing one or more of these characteristics. This directed particular attention to the last factor, because the one having the largest details and widest frets, gave the poorest results. Furthermore, when traverses were made with the 6 in. anemometer the results were considerably improved, there being a much greater difference in the reading of the two instruments than is normally the case.

In order to study the subject in a systematic manner a large number of lattice work grilles were made up out of strips of wood, approximately ⅛ in. in thickness and of various widths up to 1¾ in. and with various spacings so as to give, in each case, several amounts of free area. Traverses were made in various ways, to be discussed later, and with various types of anemometers. It was disclosed that as the size of the fret increased the error obtained by the standard formula also increased, reaching a value of over 40 per cent with a 1¾ in. fret, 30 per cent free area and the 4 in. anemometer. This means that in the case cited, it would be necessary to introduce into the formula a new coefficient having a value of approximately 0.7. It appears then that when the frets of a grille are large, it will be necessary to add a second coefficient to the normal formula, making it read

$$\text{cfm} = \frac{C_f C_v V(A + a)}{2}$$

in which C_v is the normal coefficient, recommended for grilles having small frets, and C_f a special coefficient for use only when the frets are large. Under the various circumstances this value of C_f will range from 0.65 to 1.00, being a function of three factors, namely, percentage of free area, size of fret, and size of anemometer used. These last two factors may be considered as one as it is really the ratio between the size of the fret and the diameter of the anemometer that determines the value of the coefficient, other things being equal.

The curve, Fig. 8, shows the values of C_f to be used for various widths of fret, and percentages of free area. Greater difficulty has been experienced in securing entirely consistent results in these cases than in any others dealt with. For this reason the curves are being submitted as tentative, subject to possible revision as additional data are secured. However, they may be used with the assurance that under the conditions to which they apply, the results obtained will be more accurate than would be secured by any previously recommended method. It is planned to continue the research on this point, and a set of curves in final form will be submitted as soon as they can be secured.

As an example of the application of this method, reference to the results

secured with the tests on the large plaster grille indicates that the frets of this grille varied from ¾ in. to 3 in., but a careful estimate of the average would place it at about 1½ in. The free area was 34.2 per cent. Consulting the curves it is found that the proper coefficient is 0.72. The results calculated by the normal formula were, at the higher velocities, 34 per cent too high. In other words, the calculated flow was 1.34 times as great as the actual flow. If this figure is multiplied by the new coefficient the result is:

$$\text{cfm} = 1.34 \times 0.72 = 0.965$$

or an error of 3½ per cent.

It should be emphasized again that this new coefficient need not be considered when the frets are less than ⅜ in. with the 4 in. anemometer, and ⅝ in. with the 6 in. instrument. The greater the size of the frets the more important it becomes to use great care in making the traverse. In all cases where the frets average over ¾ in. it is recommended that some one of the forms of moving traverse be used.

Methods of Making Traverse

Several methods may be followed in making the traverse to obtain the average velocity. These methods may be classified as follows:

1. Spot traverse—stationary readings.
2. Overall moving traverse.
3. Spot traverse—moving readings.
4. Strip traverse.

The first of these methods was used throughout the early part of the research, although enough readings were taken by the other methods to show that with proper care they all gave substantially the same results for the cases then under investigation. With the spot traverse method the grille face is divided into a number of equal rectangles having their least dimension approximately equal to, or greater than, the diameter of the instrument to be used. A reading of not less than 30 seconds, and preferably of one minute duration, is then taken with the anemometer held at the center of each rectangle in succession. Where the grille is very large the size of the individual rectangles may be increased so that not to exceed twenty such readings are taken. If the velocity distribution is reasonably good, it is permissible to use a smaller number of readings. In laying out these rectangles it is essential that they should not coincide with any repeating figure in the grille design except, of course, very small figures such as the squares of a small lattice work grille. For example, suppose a grille is 24 in. square and the design is made up of a six inch square pattern repeated four times in each direction. If either of the spot traverse methods are to be used, the face should be divided either into 9 squares, 25 squares or 15 rectangles each 8 x 4.8 in. This principle should be followed even though it necessitates using dimensions so small that the successive anemometer positions overlap each other slightly. The larger the detail of the pattern, that is, the wider the frets and interstices, the more important this rule becomes.

The spot traverse is disliked by many workers because of the time involved.

With these the overall moving traverse is preferred. Used with proper care this method may be very nearly as accurate as the spot method and under certain special circumstances may even become the most accurate. The usual procedure in using this method is to move the instrument along the upper edge of the grille, from one side to the other, then drop it down a short distance and return along a parallel path. This back and forth movement is continued until the entire face has been covered.

To secure satisfactory results two precautions must be observed. In the first place, the motion must be at a very steady rate, especially in those cases where the velocity changes a great deal from point to point. To maintain a uniform rate of movement it is advisable to figure out in advance exactly where the instrument should be at the end of certain time increments. For example, if it is desired to make a four minute traverse on a 2 ft square grille by making four horizontal strips across the face, the operator should keep one eye on the watch and constantly adjust his speed so that each successive 6 in. distance will be covered in 15 seconds. The second precaution is to allow for sufficient time. Here again, caution is essential in those cases involving rapid and large changes in the air velocity, whether they are due to the character of the approach or to peculiarities of the grille design. It is recommended that the instrument should be moved at a speed not greater than 2 ft per minute, and the duration of the run should be not less than *two* minutes. However, the personal element will always be a more important factor with this type of traverse than with any other, hence in general, it is not recommended where the greatest accuracy is desired.

The third method is similar to the first and differs from it only in that the anemometer is moved around slowly within the boundaries of the individual rectangles or squares. It is unnecessary to say, of course, that it is only applicable to those cases where the small rectangles are larger than the anemometer being used. With this method it is permissible to use fewer readings taken over larger rectangles. In effect this third plan is a compromise between the first two methods, and a very satisfactory one at that.

The fourth and final traverse method is really only a modification of the third. The grille is divided into parallel strips having widths approximately equal to the anemometer diameter. These must be such width, however, that they do not coincide with any repeating pattern in the grille design. The instrument should then traverse each of these strips from end to end, a separate observation being taken for each strip. If the strip is wider than the instrument it is advisable to zig-zag it back and forth as it moves along so that it follows roughly a repeating sine curve as the traverse is made.

When the grille design is made up entirely of small details and the air flow is distributed well out to the edge of the grille on all sides it makes little difference which of these methods is used. When the grille has very large details, however, carelessness in making the traverse may introduce very large errors.

As an illustration four grilles, 24 in. square, were made up of wood strips 1¾ in. wide. These four grilles consisted of 3, 4, 5 and 6 strips respectively, in each direction. Traverses were made by the first method using 16 squares. The results obtained with the 3, 5 and 6 slat grilles were consistent both with respect to each other and with respect to other similar grilles having frets of smaller size. But the four strip grille was out of line by approximately 30 per

cent, due to the fact that for each reading taken, the anemometer was in exactly the same position with respect to the strips; that is, in each case the center of the instrument was directly opposite the intersection of two frets. When the 4 in. instrument was moved about within the 6 in. square but slight change was noted. However, when nine observations were taken on 8 in. squares, and when the strip traverse was used, using 5 strips, the results were quite satisfactory. For this type of grille, and this includes the ornamental iron and plaster with large details, it is recommended that the strip traverse,

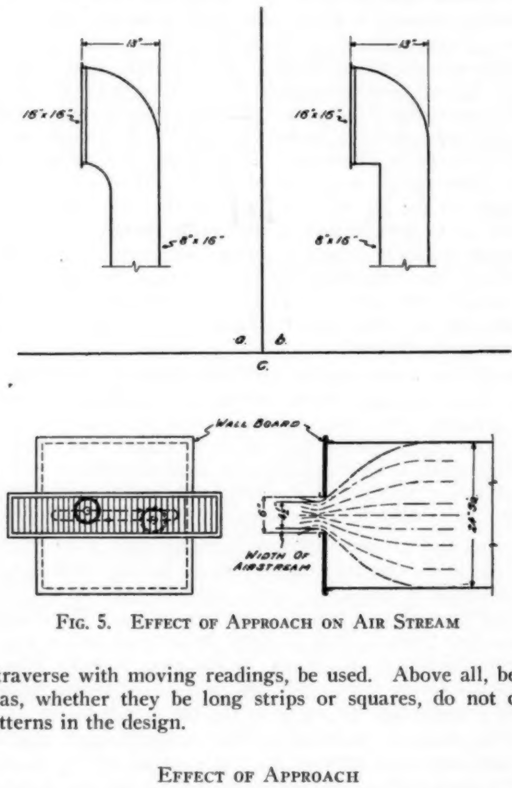


FIG. 5. EFFECT OF APPROACH ON AIR STREAM

or the spot traverse with moving readings, be used. Above all, be certain that the unit areas, whether they be long strips or squares, do not coincide with repeating patterns in the design.

EFFECT OF APPROACH

The nature of the duct as it approaches the grille will naturally affect the character of the air stream.

The approach may cause the air to be discharged at a pronounced angle to the axis of the grille, or it may cause a very bad velocity distribution. These two effects may be present either independently or simultaneously. In general, a grille possessing considerable depth in proportion to the size of the openings, such as those shown in Fig. 2, *a* and *d*, will tend to straighten the air

stream so that it will leave the grille at a 90 deg angle. They will not be particularly effective, however, in causing a uniform velocity distribution. Other things being equal the most uniform velocity distribution will be attained with a grille having a relatively small percentage of free area.

The fact that the airstream strikes the anemometer at an appreciable angle to the normal does not in itself cause any serious error until the angularity becomes extreme. E. Ower, in his book *Measurement of Air Flow,* states that an angle of 20 deg only causes an error of 1 per cent. He also reports an error of 19 per cent for 40 deg inclination. Recent experience with angular discharge from plain lattice work grilles on vertical risers showed only about 5 per cent error for a 40 deg angle of discharge. However, some portions of the air stream were at a smaller angle, which, no doubt, partly explains the relatively small error. Furthermore, the angular discharge is usually accompanied by a velocity change from a low value at the bottom to a peak value at the top. These two effects are, in part, compensating influences which also tend to reduce the error.

Considerable time was spent investigating the effect of poor velocity distribution on the accuracy of the traverse. The vertical riser shown in section *a*, Fig. 5, was one of those used to create this condition. It can be seen that it enlarges rapidly from an 8 x 16 in. stack to a 16 in. square grille. Several types of grilles were used. The discharge from the plain lattice and other thin grilles was upward at angles up to approximately 40 deg. The discharge from the grille shown in section *d*, Fig. 2, was straight forward, while from the one shown in section *b*, Fig. 3, it was downward at approximately 30 deg. These latter grilles, however, had a very large percentage of free area, and as a result the air was discharged at a high velocity at the top, while no air at all was flowing from the bottom portion. In no case was there any backflow.

The second riser, *b* in Fig. 5, had a sharp corner for the purpose of creating a reversed flow along the bottom, as this condition is encountered so frequently in practice. Although this did cause the backflow through a strip 1¾ in. wide with some of the grilles and a turbulent strip for another 2¾ in. the results, when taken by the normal method of 16 spot readings, four of which were negative, were surprisingly accurate considering the circumstances. The greatest error in any case with these two risers was 10 per cent, while the majority of the readings were accurate to within 5 per cent. However, from time to time special cases have been encountered where the error was excessive, so it is recommended that in all cases where a quiescent region or reentering air is encountered special precautions should be taken. Where there is simply a dead region this portion of the grille should be marked off and the traverse taken only over the remaining portion through which air is actually being discharged. Needless to say, only this latter part of the grille should be used in calculating the gross and net areas.

Where there is reentering air the problem is somewhat more confusing. If it is only at a very low velocity and over a small area, it may be neglected entirely, and the area treated as though it were in a quiescent state. If the amount of reentering air is large, however, it must be taken into account. To do this it is recommended that the grille face be divided into two parts, one carrying outcoming air, the other reentering air. The outcoming air should be meas-

ured by the usual method using whatever type of traverse is desired. The other portion should then be measured in the usual manner recommended for exhaust grilles. The volume of flow obtained by this second calculation should then be subtracted from the first to obtain the net volume. In many cases this procedure is not practical because the width of the strip through which the reversed flow takes place is too narrow to permit of an anemometer traverse. In such cases the best procedure will be to cover that portion of the grille with card-board or adhesive paper, so as to prevent the inflow during the time that the balance of the grille is being traversed. This method has been tried with very satisfactory results.

As has been suggested before, even if these special methods are ignored and the traverse taken by the standard procedure over the entire face the

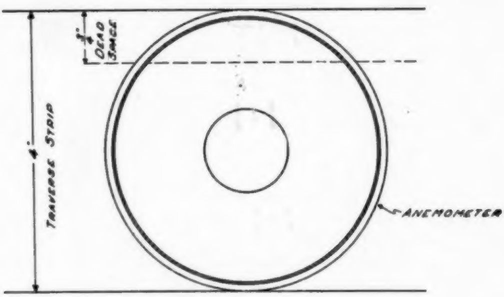


FIG. 6. EFFECT OF DEAD STRIP

results will not usually be as inaccurate as might commonly be supposed. This is only true, however, when the grille is approximately square and not too small. If one of the dimensions is quite small, say 4 to 6 in., the possibility of serious error is great because of the fact that a narrow strip along one edge may represent a very appreciable percentage of the total area. To illustrate this point a run was made with a long slender grille taken from the top of a unit ventilator. Not having an approach made to fit it exactly, it was placed across the end of a 24 in. duct and the remaining portion was blocked off with wall-board, as illustrated by *c* in Fig. 5.

Traverses were made by moving the anemometer very slowly from one end to the other while zig-zagging it up and down, and also by moving it first along the upper edge and then returning along the lower edge as indicated in the sketch. In both cases the calculated air volumes were 30 per cent too large. A careful study of the air stream was made with pitot tubes, small streamers and by holding the anemometer to one side of the grille and slowly moving it into the air stream until motion of the vanes was first observed. These observations disclosed the fact that due to the character of the approach and method of mounting, the grille was acting like any normal orifice and causing a contraction in the air stream so that the actual width of it in the plane of the anemometer vanes, was only 4½ in. When a traverse was taken using only the center strip the results were entirely satisfactory. Later another approach was obtained to fit this same grille. The contraction was then elimi-

nated and accurate results were obtained with traverses taken over the full 6 in. width.

If the dead region along one edge is of considerable width it will not usually cause serious error even though it is entirely ignored and the traverse taken by the usual method. The anemometer readings along that edge will be very low, or even negative in value, and will thus reduce the calculated average velocity by the proper amount to compensate for the dead area. When, however, the dead strip is very narrow, say 1 in. or less, it will have practically no effect in reducing the speed of the anemometer. One reason for this can be seen by studying the sketch, Fig. 6. This shows a 4 in. anemometer traversing a 4 in. strip, in which the outer ¾ in. is dead. This dead strip represents almost 20 per cent of the traverse area but only 13 per cent of the anemometer area is exposed to it. Added to this fact is the inherent charac-

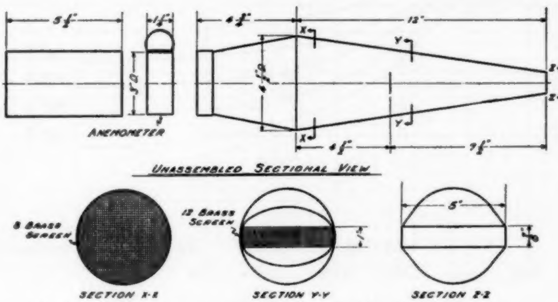


Fig. 7. New Instrument for Measuring Air Flow

teristic of anemometers, that, if there is a variation in velocity across their face, they tend to give a reading somewhat higher than the true average velocity.

Still another factor enters into the case of these slender grilles. Considering again the case of the 6 in. grille already discussed, Fig. 5, *c* shows two positions of the instrument and the course followed in making the traverse. It should be noted that the center strip, where the highest velocity is liable to exist is really counted twice and thus has a disproportionate effect upon the final indicated average velocity. In these cases, that is, narrow grilles with rapid variations in velocity from side to side, it is recommended that one of the smaller types of anemometers be used.

A narrow strip of dead space will cause more error than a wider one. Probably the worst case of all is where there is air issuing from such a narrow strip at a very low velocity relative to the remainder of the stream. A dead strip or reentering air is readily detected and can be corrected for, but a narrow low velocity strip may very easily be overlooked. It is recommended that in all cases, the edges of a grille should be carefully studied before proceeding with a traverse in order that any unusual condition may be observed and a correction made.

As strange as it may appear one of the conditions under which it is easiest to go wrong is in making a traverse across the end of an open duct. When such a traverse was made across a 24 in. duct the result was some 20 per cent

too large. This was due to the fact that with the grille resistance removed the jet coming from the 10 in. duct farther back, did not diffuse enough to completely fill the 24 in. outlet. A small dead strip, about 1 in. wide, existed all around the opening, representing 20 per cent of the outlet area and yet not wide enough to affect the anemometer readings. Placing a screen diffuser

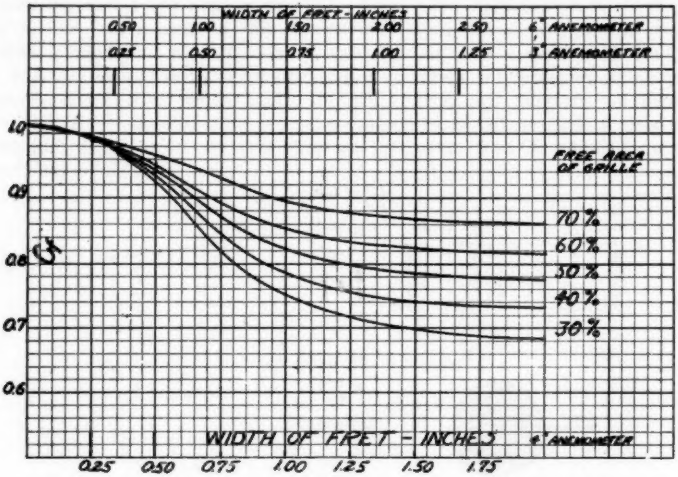


FIG. 8. CURVES SHOWING VALUES OF C_f FOR VARIOUS WIDTHS OF FRET AND PERCENTAGES OF FREE AREA

in the duct about 4 ft back of the grille corrected this condition completely and permitted a check between the anemometer and pitot traverses.

UNIT VENTILATORS

Considerable time was spent in attempting to find a satisfactory means of measuring the discharge from unit ventilators, that could be used in testing these devices in the field. For calibration purposes a long straight circular duct was connected to the suction side of each unit, as shown in Fig. 10. Pitot tube traverses were made in this entrance duct, and compared with anemometer traverses taken by various means at or near the face of the grille. Observations were made on five different types of units submitted by the various leading manufacturers.

Runs were first taken by the usual methods, both spot traverses and the various types of moving traverses. Very little difference was apparent with the different methods. When the volumes were calculated by the $\frac{V(A+a)}{2}$ formula the average results were as shown in Table 3. It can be seen that the errors were very large, although quite consistent for those runs where the air was being passed through the heater cores. It would be possible, in these cases, to introduce a coefficient of 0.73 for the 3 in. instrument, in which case

the maximum error would be about 5 per cent. This is not to be recommended, however, as it would be but a makeshift solution at best and with no assurance that it would continue to give even a moderate degree of accuracy with new designs that may be brought out in the future. In fact, Professors Larson and Nelson in their work checked the figures very closely for one unit, but differed considerably with another. The similarity in the values of the coefficient were merely due to the fact that there was a general similarity in the types of construction in the different units.

In explaining the reason for the nature of the results obtained it must be pointed out that the grilles were all of a very open character having from

TABLE 3. AVERAGE RESULTS BY CALCULATION

Ventilator	Air Delivered Through Heater Core		Air Bypassed Around Heater	
	Per Cent Error Using $\frac{V(A+a)}{2}$	Coefficient Needed to Give Accurate Results	Per Cent Error Using $\frac{V(A+a)}{2}$	Coefficient Needed to Give Accurate Results
No. 1	+31.	0.764	+58.	0.634
No. 2	+36.	0.736	+73.	0.578
No. 3	+42.	0.705	+73.	0.578
No. 4	+46.	0.688	+42.	0.703
No. 5	+29.	0.774	+43.	0.700

Notes: Results obtained with 3 in. anemometer and unheated air. Readings taken with 4 in. anemometer averaged 2 per cent more accurate.

85 to 91 per cent free area while only a short distance below them were heater cores having much smaller areas. Thus, these heaters had far more influence in determining the character of the air stream than did the grilles. Applying the principle, which has already been discussed, that the minimum area in the air stream should be used for net free area, a in the formula, the areas through the heaters should in this case be used. This method has been used with quite satisfactory results. However, the fact is recognized that it is difficult and sometimes impossible to measure or estimate accurately the free area through most heater cores. Furthermore, the flow from most of these units was quite badly distributed, especially when the air was bypassed around the heater. In fact, in the latter case, the extremely poor distribution offers the only explanation possible for the very poor results. Hence, it must be concluded that it is impractical to make direct traverses at the outlet grille of a unit ventilator with the ordinary anemometer and be able to view the results with any degree of confidence.

There has always been a tendency on the part of some experimenters to handle situations like this by placing a duct over the outlet and then traversing the open end with an anemometer. This method should always be viewed with suspicion. If the duct is straight it must be very long in order to be certain that the air stream will completely fill it. Tests were made with straight rectangular ducts, equal in dimensions to those of the outlet grille. Results obtained were 20 to 25 per cent too high although the lengths of the ducts were some eight times the least dimension of the grille. It was necessary to taper

the duct until the outlet was only about 70 per cent as great as the grille before anemometer traverses agreed with the true flow as shown by the pitot tubes. As it would be difficult to establish any fixed rule governing the amount of taper to be used, under different circumstances, and as there is always the probability that the presence of the duct will itself alter the flow, this method has never seemed to be a reasonable one.

The only solution to the problem appears to be in the development of some new device that will be relatively unaffected by non-uniform velocity distribu-

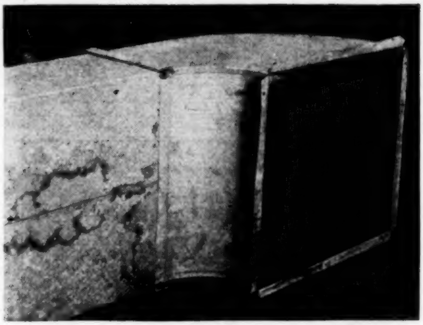


FIG. 9. SHOWING ONE TYPE OF APPROACH TESTED

tion and which will not call for a knowledge of the amount of free area. Experiments have been made with such a device, and a description of it is being included in this report.

A NEW METHOD

It is apparent that there are numerous cases where it will always be hard to measure the air flow by the usual means, due to special conditions which make it difficult or impossible to apply the rules which have been discussed. The unit ventilator represents one case as already pointed out. In general, any case where the grille proper is preceded by any obstruction of such character that it is difficult to predict its effect on the air stream will fall into this class. Another case would be that of a unit heater of the suspended type using a disc or propeller fan. Finally, grilles of a highly ornamental pattern are difficult because of the difficulty in estimating the amount of free area and the average width of the frets.

Some effort has been made to develop a new device to be used in conjunction with the anemometer that will permit making reasonably accurate measurements without any knowledge of the details of the grille other than the gross area. The objective is to be able to make a traverse with the device and then apply a simple formula:

$$\text{cfm} = K'VA$$

in which

V = indicated velocity
A = gross area of grille
K' = a coefficient—preferably a constant but possibly varying with velocity

The first attempt was a simple cylinder about 10 in. long having the same diameter as the anemometer. The theory was that the air entering this cylinder from the grille would spread out and diffuse until by the time it reached the instrument on the far end of the tube it would have a uniform velocity across the whole area of the anemometer. The variation in K proved to be too

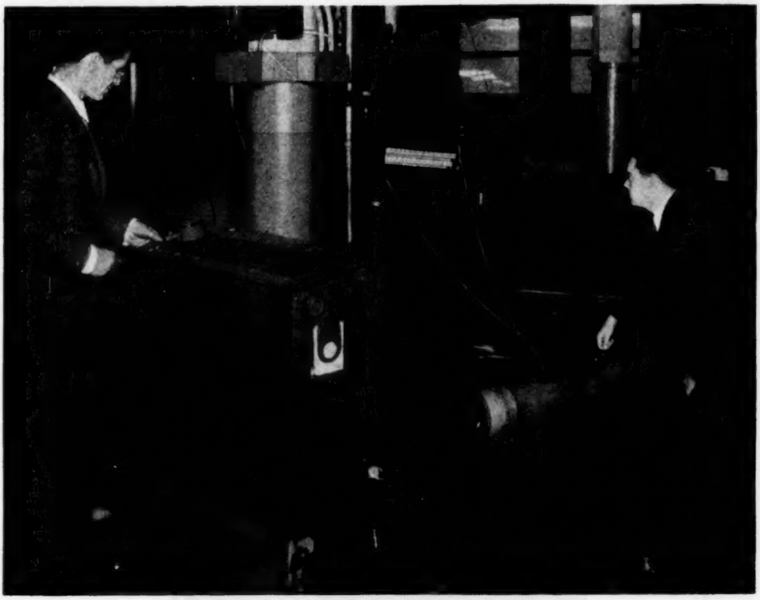


Fig. 10. Set-up for Testing Unit Ventilators

great, being rather high with grilles of large free area, and small with those having small free area. The reason for this can be seen if it is realized that the cone and anemometer introduce a resistance so that when they are held up against any given spot on a grille, less air actually passes that spot than would be the case if the device were not there. Now consider two points equal to each other in area, one on a grille having 80 per cent free area, the other having 40 per cent. Imagine, further, that the same volume of air is being delivered at both points. It follows that since the second has only half as great a free area as the first, the actual velocity of the air will be twice as great, and the kinetic energy four times as great. Naturally, this second stream will be relatively unaffected by the presence of any resistance such as the anemometer and cone.

The problem then was of a somewhat paradoxical nature. It was necessary to introduce resistance in order to break up and diffuse the jets coming from the interstices of the grille and at the same time keep the total resistance down to a minimum. After trying several types of cylinders, cones, and other special

shapes, the one shown in Fig. 7 was finally evolved. At first only the right hand double conical portion was used, with the anemometer mounted on the outer end. The cone is fitted with a handle, not shown in the sketch. Later the 3 in. tube was added, made so that it simply slips over the outer end of the anemometer. In making a traverse the grille face is divided into strips approximately 5 in. wide and the cone held with its rectangular entrance in contact with the grille and the axis at right angles to the face. The device is moved slowly from one end of the strip to the other.

It was decided that the long narrow rectangular opening would be most satisfactory in reaching the narrow dead strips along the grille edges that so

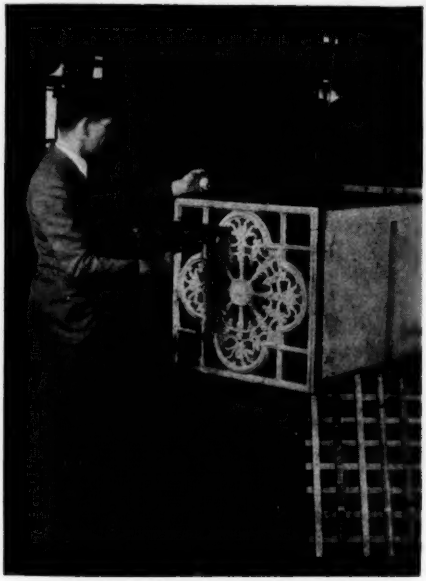


FIG. 11. SHOWING METHOD OF MAKING TRAVERSE WITH ANEMOMETER MOUNTED IN SPECIAL CONE

frequently cause trouble because of the inability of a circular instrument to reach them effectively. However, it has been found that unlike the normal anemometer traverse, the instrument is quite sensitive to the variations in the angle of flow. It is believed that if the entrance is made more nearly square this effect will be greatly reduced, so this change is to be made in the next model. It will also be practical to have two or more different types of entrance cones, each adapted to a certain class of work. At first the entrance was made of such size as to give an average coefficient of 1.00. This was a mistake and when the entrance was reduced somewhat in size, considerably greater consistency was obtained. The screens are for the purpose of breaking up the

individual air jets coming from the grille. Naturally, they introduce resistance, and it was for the purpose of partly counteracting this that the short 3 in. cylinder was added to the outlet side of the anemometer. The anemometer itself has somewhat over 20 per cent of its area occupied by the dial and supporting bars. Hence, adding a short length of 3 in. cylinder has the effect of increasing the outlet area. This, in turn, has the effect of giving a partial recovery of velocity pressure that is available for overcoming the screen resistance. The results obtained showed marked improvement when this cylinder was added. The complete device is, of course, somewhat large and slightly awkward to use, but it is thought now that it is possible to reduce the entrance cone both in diameter and length.

The instrument as illustrated has an average coefficient ranging from about 1.03 at 200 fpm to 1.15 at 800 fpm, with maximum errors of about 7 per cent. In judging the value of the instrument one must keep clearly in mind that these maximum errors occur under conditions where all other methods have failed completely to give even approximately correct results. For example, one test of particular interest was made with a suspended type unit heater. This was set against the end of a 24 in. duct in the position usually occupied by the grilles. The propeller fan was run so as to give the same velocity distribution that it would have in service, namely, a low velocity center, surrounded by a high velocity ring, and four relatively dead spots in the corners. When a traverse was made with this new device, picking the coefficient from a curve based on results obtained from several grilles covering a wide range of types, the calculated discharge from the heater was only 1 per cent in error.

Furthermore, there is every reason to believe that the accuracy of the device can be still further increased by further development. It is planned to continue these investigations and full details of the progress of the work will be furnished to the Society.

Exhaust Grilles

No additional data have been secured on exhaust grille measurements since the last report was made. The results secured at that time have been carefully studied, and a limited amount of similar data have been secured from other experimenters, as a result of which it has been concluded that the formula recommended at that time, namely

$$\text{cfm} = KVA$$

is the most accurate one possible at this time.

Most of the men applying this formula are using the average value 0.80 for K. With the exhaust grilles the air velocity had a much greater effect upon the value of the coefficient than was the case with supply grilles. For this reason, it is strongly recommended that the values of K given in The A. S. H. V. E. Guide be used for the various velocities rather than the single constant.

No observations were taken with the lattice work grilles having large frets. However, the fact that the large plaster grilles gave results in very close agreement with the average figures obtained with the small lattice designs, indicates that on exhaust the large frets do not have any material effect. However, the

usual precautions regarding the methods of making the traverse as already discussed under that heading, should be followed in order to get the best results.

DISCUSSION

JOHN HOWATT: This paper is one more example of what we are receiving through our Research Committee. Consider the papers presented at any of our meetings and you will be astounded at the large number that are a result of our research in some field. It is not always that our research brings about results that can be used in a practical way immediately. The research carried out by Professor Davies, however, is practical and gives us a handle on which we can take a hold at once.

This particular piece of research came about through a situation that arose a few years ago in Chicago. One of the most prominent ventilating contractors in that City became involved in a controversy between the representatives of the contractor and the owner as to the method of measuring the volume of air being delivered through registers of a ventilating system he installed in a large bank building. The *Ventilating Contractors Employers' Association* of Chicago took the matter up and agreed to finance the cost of research to determine the best practical field method of testing the air delivered on a ventilating job. Cooperative agreements were thereupon made between the A. S. H. V. E. and Armour Institute of Technology to carry out this work and it was so carried out under the supervision of Professor Davies of Armour Institute. The work has required 3 years of experimentation and an expenditure of over $1,000, which was contributed by the Chicago contractors.

Men of research are the pioneers in science and industry. Their work usually comes years ahead of the practical application. They open up new paths for the rest of us to follow later. Sometimes we feel that our research is so technical or so abstract, we fail to see at once the full benefits that can be derived from it because they may have no application at present. The research just completed by Professor Davies, on the other hand, is intensely practical as it can be put to use immediately and has given us the best method yet devised of making a field test of air deliveries by an air distribution system.

J. H. VAN ALSBURG: By referring to this paper we note that Professor Davies has taken into consideration practical applications which would be encountered on an installation, and he has pointed out ways and means of overcoming the usual obstacles.

Under the sub-heading, Effect of Approach, this paper states that grilles of high free air capacity usually cause a poor distribution over the face of the grille, which is largely overcome by a lesser air capacity and from this I believe we are safe to assume that an implied suggestion is made, namely, that the size of the register and the percentage of free air capacity can be used to partially control the velocity into a room. This appears to be very important in the case of small rooms.

I would also like to ask the question, namely, when the different types of grilles were tested, were any measurements taken to determine the change in total pressure or the change in volume of air delivered, as caused by the difference in grille free air capacity or grille resistance?

L. E. DAVIES: No observations were made on the changes in total pressure back of the grilles as the different types were used. As for the changes in air delivery for the different ones, with the type of set-up used this would depend as much on the fan characteristics as it would on the character of the grille.

W. W. TIMMIS: Have any studies of the air flow through concealed gravity type heaters been made in order to determine the effect of various types of construction, both with respect to the heater itself and to the enclosure and grilles?

PROFESSOR DAVIES: During the course of this investigation no observations have been made with concealed gravity type heaters.

No. 967

CARBON MONOXIDE DISTRIBUTION IN RELATION TO THE HEATING AND VENTILATION OF A ONE-FLOOR GARAGE

By F. C. Houghten † (*MEMBER*) and Paul McDermott ‡ (*NON-MEMBER*)
Pittsburgh, Pa.

THE ventilation of garages with a view of eliminating the carbon monoxide hazard with a minimum heating cost has received the attention of a Technical Advisory Committee of the American Society of Heating and Ventilating Engineers since 1928. The results of studies [1, 2] made by the Research Laboratory indicated that upward ventilation was decidedly more effective than downward ventilation. Since they were made late in the heating season when the air used for ventilation was not heated, the Laboratory was asked to determine the extent to which the same characteristics prevail when heating as well as ventilation is required.

The present study was made in the same one-story single room Sheraden Garage, described in an earlier Laboratory report.[1] Briefly, this garage has a 52-ft by 60½-ft floor space with a 11¾-ft ceiling. Windows composed a large part of the walls above the 4-ft level on three sides. It was regularly used for storage and repair purposes and during tests was occupied to about two-thirds capacity. All tests were made after midnight, when complete control of the garage was turned over to the Laboratory staff.

Test Arrangement

For the purpose of the study a central fan heating and ventilating system was installed, incorporating the usual elements of such a system, with the added characteristic of flexibility, thus making experimentation on different methods of ventilation possible. This installation is shown in Fig. 1. A fan drew outside air through a window and a heating unit and delivered it through an

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‡ Research Engr., A. S. H. V. E. Research Laboratory.

[1] Carbon Monoxide Distribution in Relation to the Ventilation of a One-Floor Garage, by F. C. Houghten and Paul McDermott, A.S.H.V.E. Transactions, Vol. 38, 1932, p. 424.

[2] Carbon Monoxide Distribution in Relation to the Ventilation of an Underground Ramp Garage, by F. C. Houghten and Paul McDermott, A.S.H.V.E. Transactions, Vol. 38, 1932, p. 439.

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Hotel Statler, Detroit, Mich., June, 1933, by J. L. Blackshaw.

overhead branch duct system to points in the center of each quarter of the garage. From these points the air could be delivered to the garage with a relatively low velocity, either obliquely upward at the 9-ft level, as shown in Sketch *A*, Fig. 2, or horizontally at the floor through a 4 sq ft opening, 4 in.

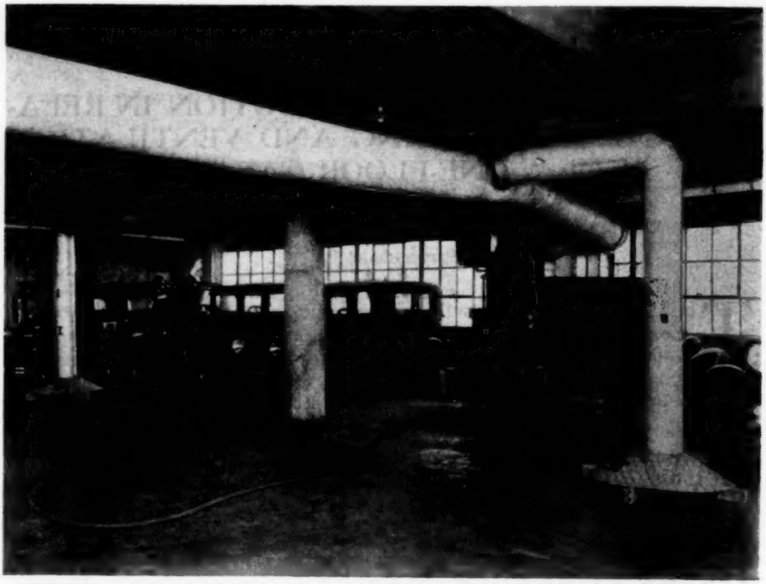


FIG. 1. INTERIOR OF GARAGE, SHOWING EXPERIMENTAL CENTRAL FAN SYSTEM ARRANGED FOR UPWARD VENTILATION

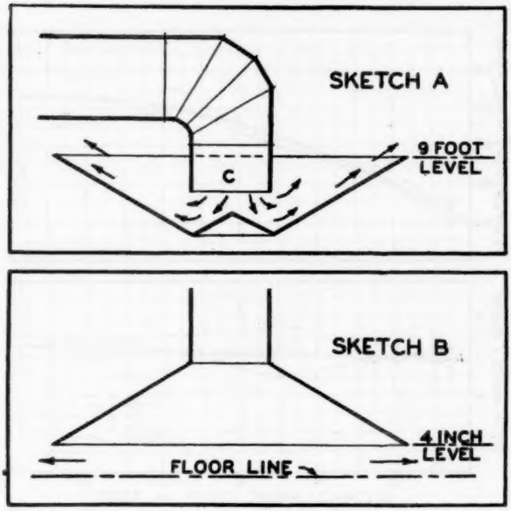
wide, around the periphery of the cone, as shown in Sketch *B*, Fig. 2. Air was removed at either the ceiling or floor level through exhaust boxes located near the four corners.

For what is termed *upward ventilation* the heated air was supplied at the floor line as in Sketch *B*, and removed through the exhaust boxes at the ceiling. For *downward ventilation* the heated air was supplied at the 9-ft level as in Sketch *A*, and removed through the exhaust boxes at the floor line. In a third type, termed *modified upward ventilation*, which was studied to a lesser extent, the deflector was removed from the end of the supply duct *C*, Sketch *A*, and the duct was extended down to the 8-ft level to allow the air to be projected from this point downward to the floor, whence it left the garage through the same exhaust box openings used for upward ventilation.

Regulation of fan speed and adjustment of the intake opening in the window allowed accurate control of the rate of air supply as indicated by a pitot tube permanently located in the main line duct. Control of distribution to the four

points of air supply was provided by a damper in each duct branch. Control of the steam supply, variations in which were noted by changes in temperature in the main line duct, permitted the maintenance of desired room temperatures.

In all tests three automobiles with idling engines were placed to give as nearly uniform distribution of exhaust gases throughout the room as possible. Special



Sketch A, downward ventilation
Sketch B, upward ventilation

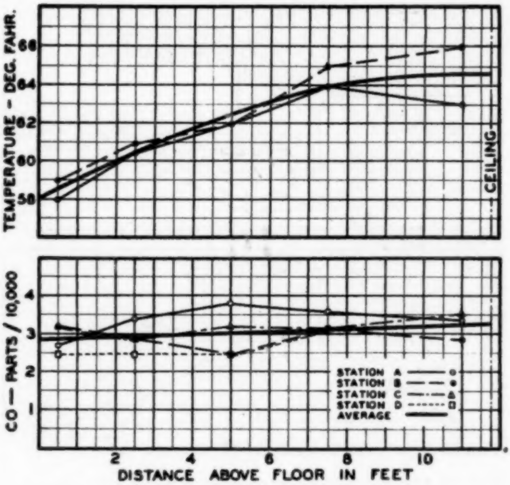
Fig. 2. Methods of Air Supply to Garage

care was taken that no exhaust pipe was directed toward any of the four sampling stations, which were spotted at representative points throughout the garage.

Preliminary to a test, the air supply and exhaust openings in the garage were adjusted for the particular type of ventilation and rate of air supply desired. The engines of the three cars were then started and the rate of steam supply was controlled to give air temperatures in the main line duct necessary to hold uniform the desired temperature at a control point 30 in. above the floor in the center of the garage. After the system had been in operation for at least an hour and constant conditions had prevailed for half an hour, samples of air for carbon monoxide analysis were taken at five elevations at each of the four sampling stations, and temperatures were observed at five elevations at two of the stations. Direct comparison of the methods of ventilation was aided by grouping into one night equal air change tests of upward, downward, and sometimes modified upward types.

TEST RESULTS

Sixteen tests were made during January and February, 1933, with outside temperatures ranging from 8 to 40 F. The results of seven tests with *upward,* five tests with *downward,* and four tests with *modified upward ventilation* are tabulated in Table 1. These tests included over 640 air samples which were analyzed for carbon monoxide. Variations found in data collected on different nights with the same type of ventilation and with the same rates of air change probably result from differences in wind velocity and direction, and in the throttle settings of the idling cars.



Data from Test *12; downward* ventilation; air change, 369,600 cu ft per hour; three idling cars; outside temperature 25.0 F

FIG. 3. CARBON MONOXIDE AND TEMPERATURE GRADIENTS FROM TYPICAL TEST

The results of a typical test are plotted in Fig. 3. Similar curves for all tests plotting carbon monoxide and temperature gradients from floor to ceiling show marked characteristics for the different types of ventilation when corrected to a common air change. In order to compare the carbon monoxide gradient curves for the three types of ventilation, they were corrected to an air change of 350,000 cu ft per hour per car. This rate of air change was accepted because an analysis of all data showed that the three idling cars gave an average rate of carbon monoxide production of 35 cu ft per hour, which with the accepted rate of air change and complete mixing gives a concentration of one part in 10,000. Incidentally, this represents 10 air changes per hour per car for the particular garage studied. The averages of the carbon monoxide

gradient curves thus corrected are plotted for the three types of ventilation in Fig. 4. The temperature gradients expressed in temperatures above that at the 30-in. level are plotted in Fig. 5 for all upward and downward ventilation tests.

Discussion of Test Results

The carbon monoxide concentration gradients, plotted in Fig. 4, indicate that at the elevation of air exhaust the same concentration existed, regardless of the method of ventilation. In other words, downward ventilation shows a ✓ concentration at the floor about equal to that shown at the ceiling by upward and modified upward ventilation. However, concentrations at the floor and throughout the lower 5 ft, or occupied region of the garage, are lower for upward ventilation than for modified upward, in which concentrations are appreciably lower than those for downward ventilation. It is obvious that the

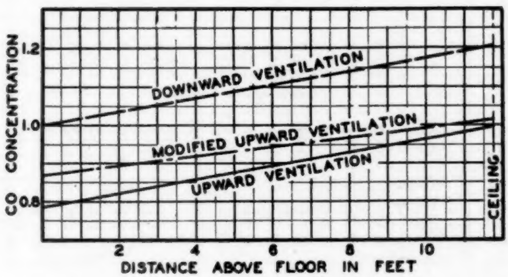


Fig. 4. Relation between Distance Above the Floor and Carbon Monoxide Concentration in Parts per 10,000 of Air. Average of All Tests Corrected to an Air Change of 350,000 Cu Ft per Hour per Idling Car

concentration at the elevation of the air removal from the garage should be the same regardless of the type of ventilation with a given air change and a given rate of carbon monoxide production. But the fact that with downward ventilation the highest concentration is found at the ceiling is not obvious, for it ✓ might be expected that with warm air supplied at the ceiling and removed at the floor a reverse gradient would be had. However, as established in the earlier Laboratory reports, the hot exhaust gases from a car rapidly rise to the ceiling, in spite of ventilation. The characteristics of the carbon monoxide gradients as found in this study indicate an increase in concentration per foot of elevation practically independent of those at the floor.

Investigations [3] by the United States Bureau of Mines and others indicate

[3] Ventilation of Vehicular Tunnels, by A. C. Fieldner, Yandell Henderson, J. W. Paul, R. R. Sayers, and others. *United States Bureau of Mines Monograph I.* (A. S. H. V. E. Journal, January-December, 1926.)

Also Bibliography in Carbon Monoxide Poisoning in Industry, Bulletin, Department of Labor, State of New York, 1930, pp. 231-238.

See also, discussion in references (1) and (2).

TABLE 1. DATA FROM CARBON MONOXIDE STUDY IN SHERADEN GARAGE

Test Number	Type of Ventilation	Air Changes		CO in Parts per 10,000 of Air				Garage Temperatures, F				Outdoor Temp, F	Rate of CO Generation per Car in Cu Ft per Hour
		Per Hour	Cu Ft per Hour	Observed		Corrected to 350,000 Cu Ft per Hour Air Changes per Car		Distance Above Floor in Ft					
				0.5 Ft Above Floor	0.5 Ft Below Ceiling	0.5 Ft Above Floor	0.5 Ft Below Ceiling	0.5	2.5	5.0	11.0		
5	Upward	4.90	171,500	5.30	5.48	0.87	0.89	59.7	60.5	61.6	64.2	34.5	31.15
8	Upward	4.60	161,000	4.45	7.28	0.68	1.11	59.4	60.4	61.8	65.2	24.5	38.85
11	Upward	10.94	382,900	2.82	3.91	1.03	1.43	61.6	62.3	63.1	65.0	24.5	50.05
14	Upward	10.15	355,250	2.60	2.91	0.88	0.98	58.2	59.1	60.1	62.5	9.8	34.30
16	Upward	9.96	348,600	2.50	2.80	0.83	0.93	70.9	71.8	73.0	75.8	39.5	32.55
17	Upward	5.45	190,750	2.70	2.75	0.74	0.75	71.0	72.0	73.2	76.1	39.5	26.25
18	Upward	5.45	190,750	2.00	2.85	0.55	0.78	70.4	71.2	72.4	75.3	39.5	27.30
Average						0.797	0.981						34.35
2	Downward	5.53	193,550	7.45	7.90	1.37	1.46	64.8	69.2	73.1	75.8	37.0	47.95
6	Downward	4.83	169,050	4.60	5.08	0.74	0.82	55.3	59.3	63.3	67.2	32.0	25.90
7	Downward	4.92	172,200	5.00	4.30	0.82	0.71	57.0	60.0	63.3	67.4	25.3	28.70
12	Downward	10.56	369,600	2.91	3.22	1.02	1.14	58.5	60.4	62.4	64.6	25.0	35.35
13	Downward	9.95	348,250	3.30	5.50	1.09	1.82	56.0	60.0	63.7	66.5	8.5	38.15
Average						1.006	1.190						35.28
4	Mod. Up.	4.85	169,750	6.70	6.18	1.08	1.00	58.9	59.5	60.2	61.7	35.0	35.00
9	Mod. Up.	4.51	157,850	5.20	6.10	0.78	0.92	62.5	62.7	62.9	63.5	24.5	32.20
10	Mod. Up.	10.74	375,900	1.82	3.20	0.65	1.15	59.7	60.0	60.3	61.2	24.0	40.25
15	Mod. Up.	9.86	345,100	3.00	2.90	0.99	0.95	62.4	62.2	61.9	61.1	8.0	33.25
Average						0.875	1.005						35.18

that men should not be subjected to concentrations of more than 4 parts of carbon monoxide in 10,000 parts of air, and that no atmosphere with concentrations over one part in 10,000 should be considered safe for extended periods of time.

The concentrations of carbon monoxide found in the different tests with different rates of air change give rates of carbon monoxide production per

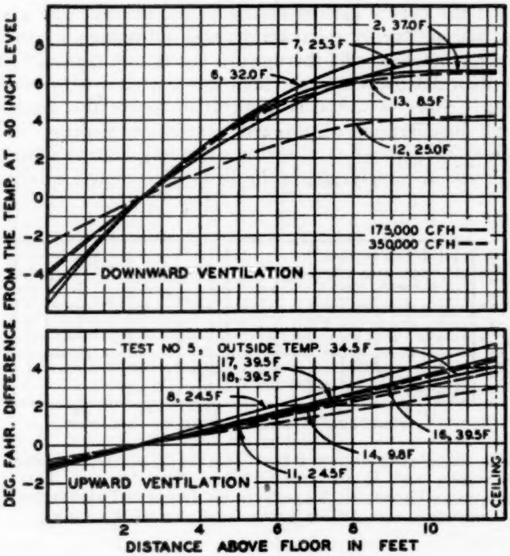


Fig. 5. Temperature Gradients from Floor to Ceiling Expressed in Degrees Difference from the Temperature Control Point, 30 In. Above the Floor

idling car ranging from 25.9 to 50.1, with an average rate of 35 cu ft per hour per car for all tests here reported, which compares reasonably well with rates of from 27.6 to 34.8 found for the earlier Laboratory studies and with the rates of 31.0 to 39.2 for passenger cars and light trucks published[4] by the United States Bureau of Mines. Accepting 35 cu ft per hour, or the average rate of carbon monoxide production for all the tests made in this study, as a basis, calculations show that an air change of 350,000 cu ft per hour per idling car is necessary to give a concentration of one part in 10,000 when there is complete mixing of exhaust gases with air supplied. The curves in Fig. 4 are based upon this rate of air change. They indicate that in order to keep the concentration of carbon monoxide down to any accepted limit at an elevation of 5 ft, approximately

[4] Ventilation of Garages, by G. W. Jones and S. H. Katz. A.S.H.V.E. Transactions, Vol. 29, 1923, pp. 341-346.
See also, references (1) and (2).

24 per cent more air must be supplied with downward ventilation than with upward ventilation. In other words, with one idling car an air change of 306,000 cu ft per hour is necessary to keep the concentration down to one part of carbon monoxide in 10,000 parts of air at the 5-ft level with upward ventilation, as compared to an air change of 380,000 cu ft per hour needed to keep the concentration down to the same limit with downward ventilation.

Considering the vicious hazard incident to exposure to carbon monoxide, and the fact that the curves plotted in Fig. 4 are averages of values showing considerable variation, one is reluctant to take advantage of the smaller air change, indicated as necessary by the upward ventilation curves, by using an air change of less than 350,000 cu ft per hour per car shown by computation for complete mixing. Since it would be difficult to insure the attainment of this advantage in all installations of upward ventilation systems, it would seem desirable to allow it to stand as a factor of safety and to ventilate with the upward method, and to supply air on the basis of complete mixing. It should

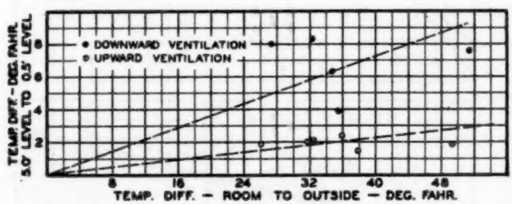


FIG. 6. RELATION BETWEEN TEMPERATURE DIFFERENCE BETWEEN THE ROOM AND OUTSIDE AIR AND THE TEMPERATURE GRADIENT, EXPRESSED IN DEGREES DIFFERENCE, BETWEEN THE 5.0-FT LEVEL AND THE 0.5-FT LEVEL

be emphasized, however, that if this basis is accepted, a factor of safety of approximately 12.5 per cent can be expected with upward ventilation at the 5-ft level, while downward ventilation fails to give as good conditions as those resulting from complete mixing by 8.5 per cent.

Inspection of the temperature gradient curves in Fig. 5 shows rather definite relationships between the temperature gradients and rates of air change and outside temperature. The points for upward and downward ventilation in Fig. 6 were obtained by plotting the temperature difference between the 0.5-ft level and the 5-ft level against the temperature difference between the room control point and the outside. While the points are too scattered to allow the drawing of definite curves, they show a tendency for increased temperature gradient with increased temperature difference between the outside and inside, and a pronounced separation for the two types of ventilation.

Plotting the temperature gradient, expressed in temperature difference between the 5-ft level and the 0.5-ft level, against rate of air change gives the points in Fig. 7. While these points are rather scattered, they indicate a

definite tendency toward a decreased temperature gradient with increased air change, and also a definite separation for the two types of ventilation.

The temperature gradient curves in Fig. 5 are steeper, particularly through the lower half of the room, with downward ventilation than with upward ventilation. The falling off of this gradient at higher elevations with downward ventilation probably results from the fact that the heated air was discharged with good distribution and mixing at the 9-ft level rather than at the ceiling. Since, with the same control temperature at the 30-in. level, the downward ventilation gives a higher air temperature next to the ceiling and upper part of the side walls, the heat loss from this part of the exposure and therefore for

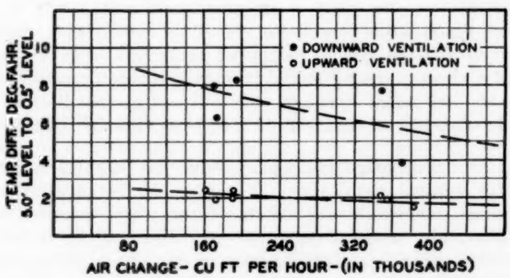


FIG. 7. RELATION BETWEEN AIR CHANGE AND TEMPERATURE GRADIENT, EXPRESSED IN DEGREES DIFFERENCE, BETWEEN THE 5.0-FT LEVEL AND THE 0.5-FT LEVEL

the entire garage must be greater. Hence, downward ventilation gives a less efficient heating effect.

No doubt, a heat balance made in conjunction with the study would have shown a considerably higher rate of heat input with downward ventilation than with upward ventilation for the same night, the same air change, and the same control temperature at the 30-in. level. Unfortunately, the demands on the garage would not permit running tests for a sufficient length of time to establish a constant and measurable rate of heat input, measured either by the rate of steam consumption or by the duct temperatures. While the duct temperatures do not give sufficiently consistent results to allow the development of a fixed percentage difference in heat input, they were consistently lower with upward ventilation. While the greater efficiency resulting from the better heating effect of upward ventilation cannot be numerically evaluated from the data presented, it must nevertheless be considerable, and when added to the possible saving in heat due to the lower air change leaves no doubt concerning the superiority of the upward system of ventilation.

Probably in a garage more than in any other type of workshop, a uniform temperature from the floor to the breathing line is desirable, because men

frequently must work under cars. Hence, because upward ventilation gives more uniform temperatures near the floor, it has an advantage in addition to the savings in heating load.

PRACTICABILITY AND EFFICIENCY OF THE UPWARD AND MODIFIED UPWARD SYSTEMS OF VENTILATION

The particular shape of the cone outlet used with upward ventilation and the shape of the deflector used with downward ventilation should not be confused as a requirement in either of these systems. They represented a convenient way of obtaining the result desired, namely, upward oblique distribution of air near the ceiling in case of downward ventilation, and low velocity horizontal distribution of air over the floor in upward ventilation. Any other shape or arrangement of outlet satisfying these essentials should give identical results. In the upward ventilation system the outlet succeeded in giving an air stream in the form of a sheet reaching to the boundary of each quarter of the garage supplied by each individual opening. In a practical installation this requirement should be considered essential in attaining similar results.

Modified upward ventilation indicated results only slightly less effective than those shown for upward ventilation. It is possible that application will prove this method more practicable, regardless of the slight superiority of upward ventilation. Certainly, modified upward ventilation can be installed with less difficulty and will offer less interference in the practical use of a garage. However, it is probable that contingent factors met with in practice would result in less certainty that the results found in this study would be obtained with modified upward than with upward ventilation. Obviously, the outlets with modified upward ventilation must be so designed that the air stream will carry to the floor and spread. A single car standing below an opening would defeat the effectiveness of such a system.

SUMMARY AND CONCLUSIONS

1. Upward ventilation results in a lower concentration of carbon monoxide at the breathing line and a lower temperature above the breathing line than does downward ventilation, for the same rate of carbon monoxide production, air change and the same temperature at the 30-in. level.

2. A lower rate of air change and a smaller heating load are required with upward than with downward ventilation.

3. Upward ventilation results in more uniform temperatures throughout the occupied portion of a building, a condition particularly desirable in a garage where men work close to the floor.

4. The study indicates that in the average case upward ventilation results in a lower concentration of carbon monoxide in the occupied portion of a garage than is had with complete mixing of the exhaust gases and the air supplied. However, the variations in concentration from point to point, together with the possible failure of the advantages of upward ventilation to

accrue, suggest the basing of garage ventilation on complete mixing and an air change sufficient to dilute the exhaust gases to the allowable concentration of carbon monoxide.

5. The rate of carbon monoxide production by an idling car is shown to vary from 25 to 50 cu ft per hour, with an average rate of 35 cu ft per hour. This is in agreement with earlier Laboratory studies and the findings of the United States Bureau of Mines.

6. Accepting an average rate of carbon monoxide production of 35 cu ft per hour per idling car, an air change of 350,000 cu ft per hour per idling car is required to keep the carbon monoxide concentration down to one part in 10,000 parts of air.

DISCUSSION

E. K. Campbell: The results of this research are quite conclusive in showing that the economical way to ventilate a garage from the standpoint of carbon monoxide is to discharge the heated air so that it spreads across the floor.

In discussing this problem in the committee, it was pointed out that, if the air were introduced horizontally, it would tend to rise and would dilute largely the upper layers of the air where no men are at work. Therefore it was decided to try introducing the air with a downward movement of considerable force so that the heated air would be driven clear to the floor and would spread as it struck the floor. This was modified in this particular test by carrying the ducts clear to the floor, but it is interesting to note that there is comparatively little difference in results between carrying the ducts to the floor and discharging the air at about 8 ft above the floor. Dropping the duct down to the floor would of course not be practicable under ordinary conditions. It was therefore valuable information to find that it is not necessary in order to get the proper results.

I am inclined to disagree with the authors of the paper in their conclusion that any saving effected by this method should be reserved as a safety factor, and that the amount of air required should be the theoretical amount of air required to dilute the carbon monoxide thrown off by the average engine, assuming equal distribution throughout the entire space. That is an easy solution of the problem and it would be easy to develop a code that would specify a certain volume of air in proportion to certain cars, etc., but the problem that confronts the committee is to get a code that can be applied by the ordinary garage owner and operator without too great hardship.

There is considerable leakage to take into account. It all helps to dilute the carbon monoxide. Furthermore the leakage helps to dilute the carbon monoxide in the lower layers of the air. In the interest of economy and in the interest of having the code applied in actual practice, it should not be ignored.

The committee has underway in Kansas City, a field test handling an actual problem as it occurred in a garage. The report could not be prepared in time for presentation at this meeting, but a study is being made of a method of using the infiltration through opening doors and the effect that its proper use will have on the volume of

air required to be supplied through the ventilating heaters. We have reached the conclusion that it is just as important to supply additional heat as it is to supply additional air from out of doors. We have noticed that because of the additional heat we are putting in the garage under study, the operators are inclined to leave the doors open longer, consequently the infiltration through the big open doors tends to sweep the carbon monoxide out of the building and relieve the condition with a comparatively small volume of air in proportion to the total cubical contents.

The committee has considerable work to do to complete this study, but we should bear in mind that whatever our final results may be, the operating costs must be within practical limits and the reach of the ordinary garage operator, else they are of little value.

In Memoriam

Names	Joined the Society	Died
Frank I. Cooper	1911	Oct. 1933
Andrew C. Edgar	Charter Member	Feb. 1933
Richard Hankin	1898	Nov. 1933
Hosford D. Kellogg	1916	Jan. 1933
Frank G. McCann	1903	May 1933
Raleigh D. Morrill	1928	Mar. 1933
William A. Pope	1906	Feb. 1933
John E. Ryan	1931	Sept. 1933
John R. Shanklin	1899	June 1933
William J. Smallman	1911	Sept. 1933

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